ECE 445

SENIOR DESIGN LABORATORY

FINAL REPORT

Movable Impact Testing Platform

Team #445

YIHANG SHEN
(yixing3@illinois.edu)
SHANGYU WANG
(shangyu3@illinois.edu)
BINGKUN FU
(pingkun3@illinois.edu)
FEIYU TANG
(feiyut3@illinois.edu)

TA: Xiangpu Jia
Sponsor: Prof. Binbin Li

May 15, 2026

Abstract

The Movable Impact Testing Platform is a vehicle-mounted electro-hydraulic prototype designed to generate repeated ground-contact impact motion. The system uses a $72\ \mathrm{V}$ battery as the main power source. The high-power path drives a brushless DC motor and hydraulic pump through a main switch or air circuit breaker and a DC inverter/motor drive, while separate DC-DC converters provide $24\ \mathrm{V}$ power to the servo proportional directional valve and $5\ \mathrm{V}$ power to the microcontroller and printed circuit board (PCB) control circuit. The control circuit generates a fixed-frequency $\pm 10\ \mathrm{V}$ command signal for the valve, which changes the hydraulic flow direction and drives repeated hydraulic cylinder motion. The final prototype was assembled on the vehicle platform and verified through control-signal, power-supply, and integrated-motion tests. The battery measured $78.0\ \mathrm{V}$ with no load and $77.8\ \mathrm{V}$ during operation, while the valve and control supplies measured $24.1\ \mathrm{V}$ and $5.0\ \mathrm{V}$, respectively. Video-based cycle counting showed 21 complete reciprocating cycles in $5.0\ \mathrm{s}$, corresponding to a measured output frequency of $4.2\ \mathrm{Hz}$. These results demonstrate that the prototype successfully converts an electrical valve command into repeated hydraulic impact-end motion, although further force, pressure, and long-duration reliability testing would be required before field deployment.

Contents

1 Introduction

Ground-contact drilling and impact mechanisms are used in many field applications, including shallow pavement evaluation, soil testing, geotechnical investigation, and material penetration tasks. In these applications, the testing or drilling system must often be moved between different locations rather than operated only at a fixed laboratory station. For example, the Dynamic Cone Penetrometer (DCP) is used to assess the in situ strength of undisturbed soil and compacted materials in shallow pavement applications [1]. Larger truck-mounted drilling rigs are also used for geotechnical and geoenvironmental site investigation, including coring, in situ testing, sample collection, and borehole instrumentation [2]. These examples show the practical value of mobile systems that can apply repeated mechanical contact to ground or pavement surfaces.

Existing field drilling and impact systems cover a wide range of sizes and capabilities. Manual devices are relatively portable, but their output depends strongly on operator consistency and they provide limited electronic control. Commercial drilling rigs can provide large force and reliable field operation, but they are expensive, heavy, and far beyond the scale of a compact senior design prototype. Laboratory hydraulic test systems can produce controlled motion, but they are usually stationary and require dedicated support equipment. This project is motivated by the gap between these two categories: a need for a compact, movable, and electronically controlled mechanism that can demonstrate repeated ground-contact motion without requiring a full-scale industrial drilling platform.

Hydraulic actuation is a suitable approach for this type of system because it can provide large linear force using a compact actuator. Prior work on hydraulic hammer drilling shows that repeated hydraulic impact can be useful in drilling and penetration applications [3]. In hydraulic systems, directional control valves are commonly used to change fluid flow direction from a hydraulic power unit and actuate hydraulic cylinders or motors **grainger˙directional˙valves**. These characteristics make an electro-hydraulic architecture appropriate for a vehicle-mounted drilling or impact prototype: the hydraulic subsystem provides force, while the electrical subsystem provides command and timing control.

This project develops the Movable Impact Testing Platform, a vehicle-mounted electro-hydraulic system for repeated ground-contact impact motion. The prototype combines a battery-powered hydraulic supply, a motor-driven pump, a servo proportional directional valve, a hydraulic cylinder, an impact or drill-bit actuation structure, and a microcontroller-and-PCB-based control circuit. The high-power portion of the system supplies hydraulic pressure and drives the actuator, while the low-power control portion generates the valve command that determines the motion sequence. The final system is intended to be mounted on a vehicle so that the impact mechanism can be positioned at different test locations.

The main engineering goal of the project is to demonstrate a working electrical-to-hydraulic-to-mechanical conversion chain. Electrical power from the 72 V battery is routed through a main switch or air circuit breaker and then through a DC inverter/motor drive, which converts the battery output into a three-phase motor drive signal for the brushless DC motor. The motor drives the hydraulic pump, and the pump generates the hydraulic pressure required by the actuator. The control circuit then commands the servo proportional directional valve, which changes the hydraulic flow path. The hydraulic cylinder converts the controlled hydraulic flow into linear motion, and the mechanical structure transfers that motion to the impact or drill-bit end.

Figure 1 shows the overall system organization. The diagram should be read as a high-level block diagram rather than a detailed wiring or hydraulic schematic. The detailed design of each subsystem is presented in the Design section.

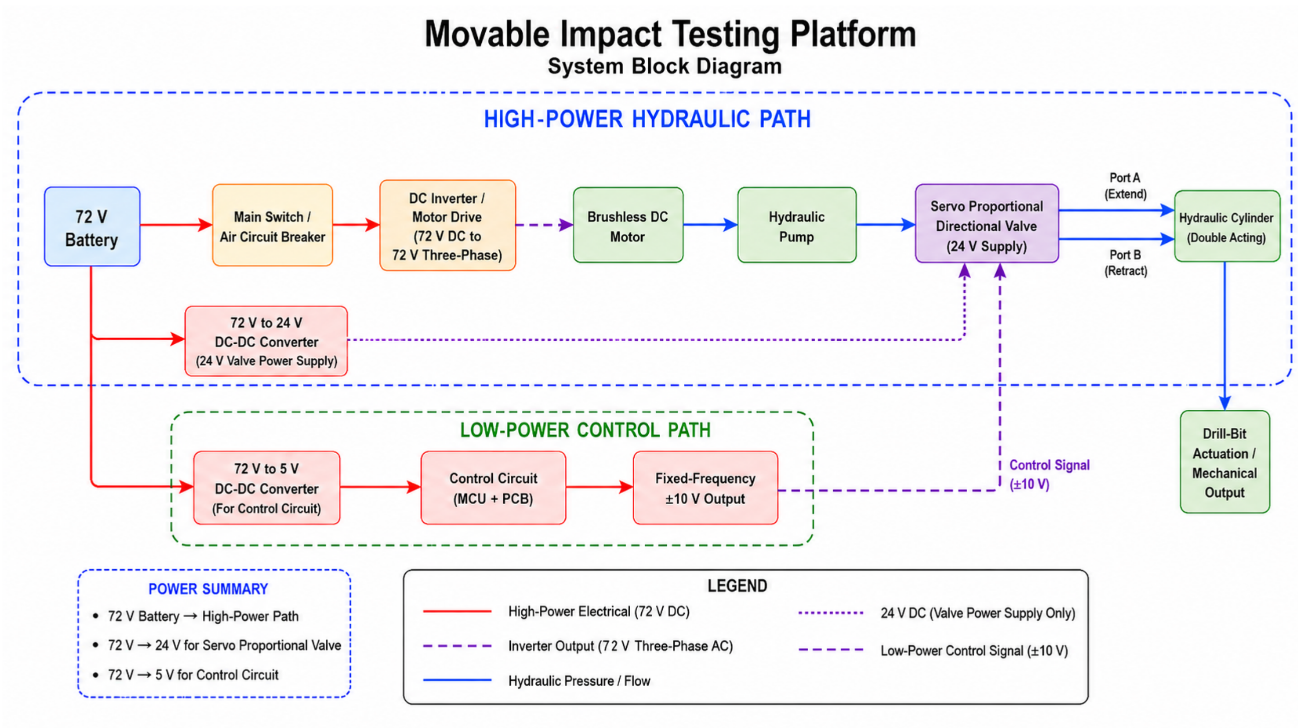

Figure 1: System-level block diagram of the Movable Impact Testing Platform. The high-power path supplies hydraulic pressure, the 72 V to 24 V converter powers the servo proportional directional valve, and the low-power control path generates the fixed-frequency ± 10 V valve command.

The rest of this report describes the design, verification, cost, and conclusions of the prototype. The Design section explains the major subsystems, including the power supply, motor drive, hydraulic pump, valve control, cylinder actuation, and vehicle-mounted mechanical integration. The Verification section presents the main tests used to confirm that the electrical command path, hydraulic supply path, and mechanical output operate as intended. The Cost section summarizes labor and component costs. The Conclusion section discusses the accomplishments, limitations, safety considerations, and possible future improvements of the project.

2 Design

2.1 System Overview

The project is designed as a vehicle-mounted electro-hydraulic drilling and impact mechanism. The system converts electrical energy from an onboard battery into hydraulic pressure and then converts the hydraulic energy into repeated linear motion at the drill-bit end. The key design feature is that the mechanical output is not controlled by directly driving the hydraulic cylinder with an electric actuator. Instead, the cylinder direction is controlled through voltage-based switching of an electro-hydraulic directional valve. This architecture separates the high-power hydraulic actuation path from the low-power electronic control path.

The complete system can be divided into six major functional blocks: the power supply and motor drive, the hydraulic pump and pressure supply, the voltage-based valve switching control, the servo proportional directional valve, the hydraulic cylinder and impact-end actuation assembly, and the vehicle-mounted mechanical structure. The 72 V battery supplies three electrical branches. The high-power branch passes through the main switch or air circuit breaker and the DC inverter/motor drive to power the brushless DC motor. A second branch uses a 72 V to 24 V DC-DC converter to power the servo proportional directional valve. A third branch uses a 72 V to 5 V DC-DC converter to power the microcontroller and PCB control circuit. The hydraulic pump generates pressurized oil, while the control circuit generates a fixed-frequency ± 10 V command signal for the valve. The valve changes the hydraulic flow direction, causing the hydraulic cylinder to extend and retract.

The main design objective is to generate a repeated drill-bit motion at a controlled frequency while maintaining enough hydraulic force for ground-contact operation. The final control strategy uses a 10 Hz voltage switching command to control the directional valve. Because one complete mechanical cycle of the drill bit requires two valve states, one for the extension stroke and one for the retraction stroke, the resulting drill-bit reciprocation frequency is 5 Hz. This relationship is central to the system design and is expressed as

$$f_{\text{drill}} = \frac{f_{\text{switch}}}{2} \tag{1}$$

where f_{switch} is the electrical voltage switching frequency and f_{drill} is the resulting mechanical reciprocation frequency at the drill bit. For the implemented prototype,

$$f_{\text{drill}} = \frac{10\ \text{Hz}}{2} = 5\ \text{Hz} \tag{2}$$

Equation 2 shows why the measured or expected drill-bit frequency is half of the valve-command switching frequency. In this design, the 10 Hz signal does not represent ten complete drill-bit cycles per second. It represents ten direction commands per second.

A complete drill-bit cycle consists of one forward motion and one return motion, so the output frequency is $5\ \mathrm{Hz}$.

The system overview is shown conceptually in Figure 1. The high-power hydraulic path starts from the $72\ \mathrm{V}$ battery, passes through the main switch or air circuit breaker, and enters the DC inverter/motor drive. The inverter converts the $72\ \mathrm{V}$ DC input into a $72\ \mathrm{V}$ three-phase motor drive signal for the brushless DC motor. The motor drives the hydraulic pump, and the pump supplies pressurized hydraulic fluid to the servo proportional directional valve and hydraulic cylinder. In parallel, a $72\ \mathrm{V}$ to $24\ \mathrm{V}$ DC-DC converter powers the servo proportional directional valve, while a separate $72\ \mathrm{V}$ to $5\ \mathrm{V}$ DC-DC converter powers the microcontroller and PCB control circuit. The control circuit generates the fixed-frequency $\pm 10\ \mathrm{V}$ valve command that determines the hydraulic flow direction.Figure 2 shows the final vehicle-mounted prototype after the major electrical, hydraulic, and mechanical modules were assembled on the vehicle frame.

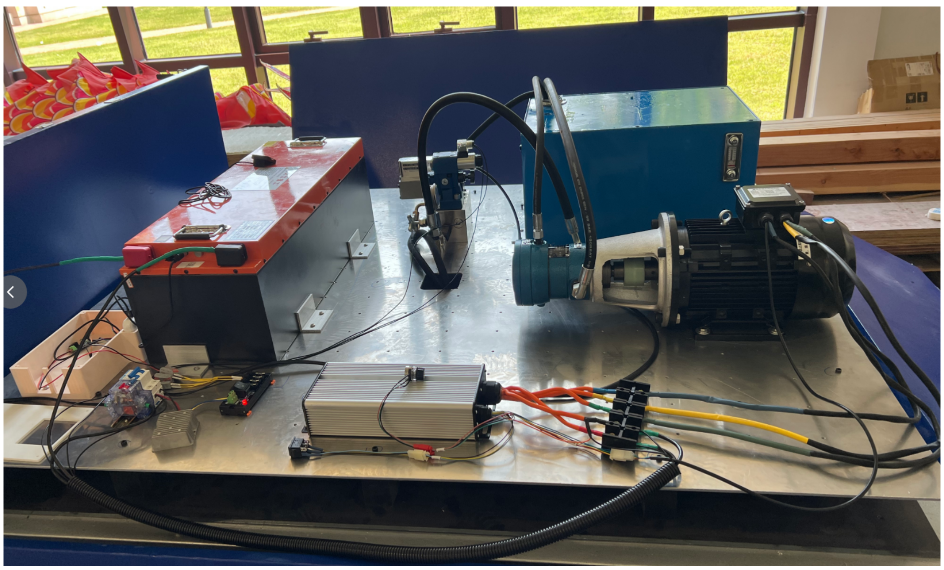

Figure 2: Final vehicle-mounted prototype of the Movable Impact Testing Platform. The platform integrates the battery, motor drive, hydraulic pump and reservoir, servo proportional directional valve, hydraulic cylinder, control electronics, and mechanical mounting structure on the vehicle frame.

This division of the system also simplifies the engineering design process. The hydraulic components are selected based on pressure, flow, force, and mechanical robustness. The electrical control components are selected based on voltage compatibility, switching frequency, current capacity, and reliability. The mechanical mounting structure is designed to keep the drill-bit motion aligned with the intended contact direction after the system is installed on the vehicle. As a result, each subsystem has a clear function, but all subsystems contribute to the final $5\ \mathrm{Hz}$ drill-bit output.

2.2 Power Supply and Motor Drive

The power subsystem provides energy for three electrical loads: the pump motor drive, the servo proportional directional valve, and the microcontroller-and-PCB-based control circuit. The pump motor is the dominant load because it provides the mechanical power needed to maintain hydraulic pressure and flow. The servo proportional valve requires a separate 24 V supply for valve operation. The control circuit requires a regulated 5 V supply so that the microcontroller and PCB can generate a repeatable fixed-frequency ± 10 V command signal.

The prototype uses a 72 V battery as the primary energy source. The battery output is routed through a main switch or air circuit breaker before reaching the high-power motor-drive path. This switch provides a simple way to enable or disable the main power path and protects the system during setup and testing. After the switch, the high-power path enters the DC inverter/motor drive, which converts the 72 V DC input into a 72 V three-phase drive signal for the brushless DC motor.

The DC inverter/motor drive controls the electrical power delivered to the brushless DC motor. Instead of connecting the 72 V battery directly to the motor, the drive converts the DC battery input into a three-phase motor drive output. This conversion is necessary because the pump motor is driven as a three-phase motor rather than as a simple two-terminal DC load. By controlling the motor through the inverter, the system can operate the hydraulic pump in a more controlled and reliable manner during testing.Figure 3 shows the DC inverter/motor drive and the three-phase motor wiring used in the high-power path.

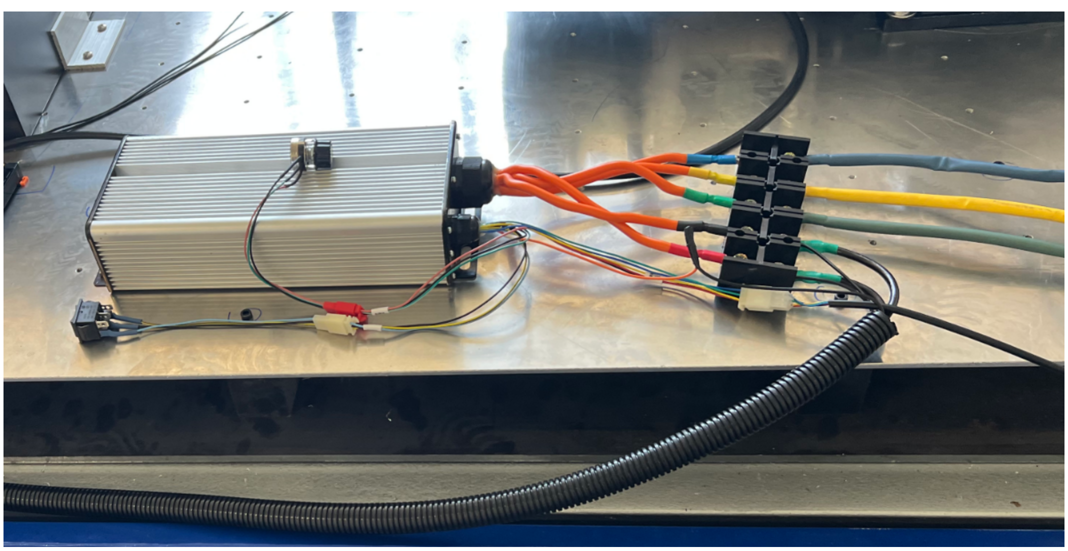

Figure 3: DC inverter and motor-drive module used to drive the brushless DC motor from the 72 V battery supply. The three-phase output wiring connects the drive to the pump motor.

The power design therefore has four main requirements. First, the 72 V battery must supply enough power for the motor drive and auxiliary power converters. Second, the DC inverter/motor drive must be compatible with the 72 V battery input and must provide the required three-phase output for the brushless DC motor. Third, the 72 V to 24 V DC-DC converter must provide stable power to the servo proportional directional valve. Fourth, the 72 V to 5 V DC-DC converter must provide stable power to the microcontroller and PCB control circuit.Figure 4 shows the assembled power and control module, including the converter and control hardware used to supply the servo valve and the microcontroller-based control circuit.

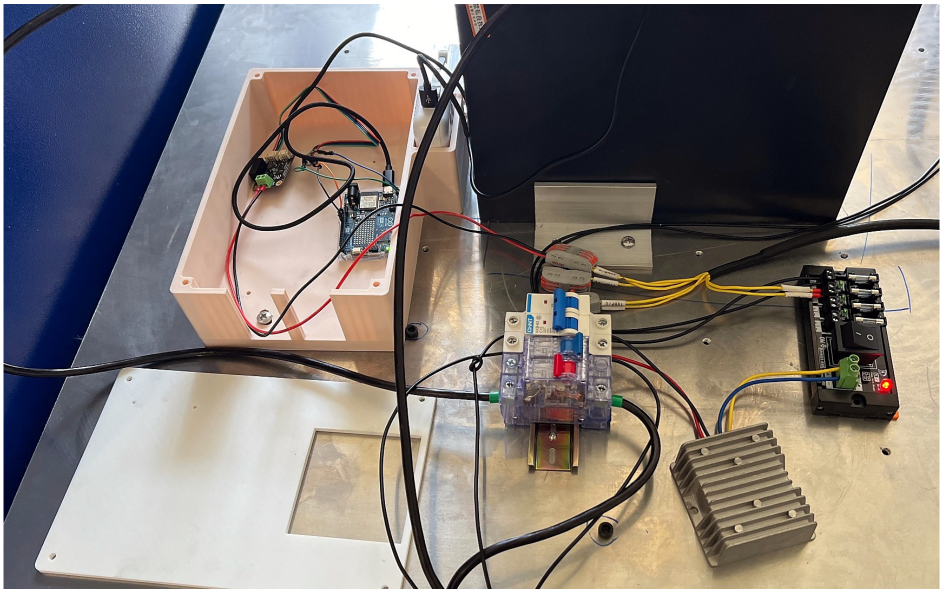

Figure 4: Power and control module mounted on the vehicle platform. The module includes the 72 V battery connection, main switch or air circuit breaker, DC-DC converters, microcontroller-and-PCB control circuit, and wiring terminals for the valve and control outputs.

Table 1: Power subsystem design quantities to be verified

Quantity	Design Value	Verification Method
Battery voltage	72 V nominal	Multimeter measurement
Motor drive output	72 V three-phase	Motor drive output measurement
Servo valve power supply	24 V DC	DC-DC converter output measurement
Control circuit supply	5 V DC	DC-DC converter output measurement
Valve command signal	Fixed-frequency ± 10 V	Oscilloscope measurement

The motor drive also affects the hydraulic performance. Hydraulic pump output is related to pump speed, and pump speed is controlled by the motor. If the motor cannot maintain adequate speed under load, the hydraulic pressure or flow may decrease, which would reduce cylinder motion speed and drill-bit impact consistency. For this reason, the motor drive is treated as part of the hydraulic performance chain rather than as an isolated electrical component.

The battery, main switch or air circuit breaker, motor drive, DC-DC converters, and control PCB should be mounted in fixed locations so that their weight and wiring do not shift during vehicle movement or impact operation. These mechanical and safety considerations are part of the power design because an electrical failure in this project could lead to both loss of control and hydraulic motion hazards.

2.3 Hydraulic Pump and Pressure Supply

The hydraulic pump and pressure supply subsystem converts the electrical output of the motor drive into pressurized hydraulic fluid. This subsystem includes the pump motor, hydraulic pump, reservoir, hoses, fittings, and pressure-control components. Its function is to provide the pressure and flow required for the directional valve and hydraulic cylinder. Figure 5 shows the motor-pump assembly used to generate hydraulic pressure for the actuation system.

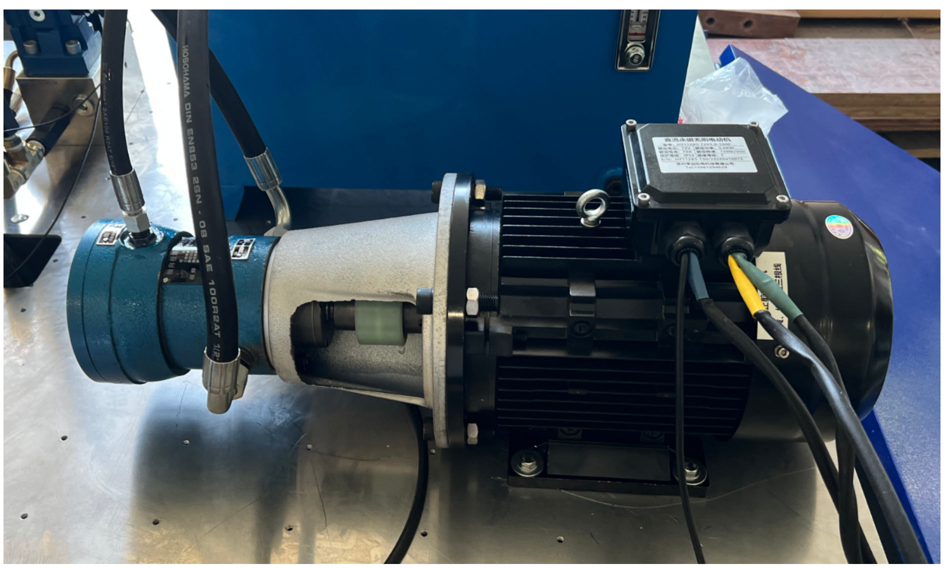

Figure 5: Brushless DC motor and hydraulic pump assembly. The motor converts electrical drive power into mechanical rotation, and the pump converts the motor output into hydraulic pressure and flow.

Hydraulic actuation was selected because the drilling mechanism requires a larger linear force than a compact electric actuator can easily provide at the same size and cost. A hydraulic cylinder can generate high force from moderate pressure because the output force is proportional to the piston area. The basic force relation is

$$F = PA \tag{3}$$

where F is the cylinder output force, P is the hydraulic pressure, and A is the effective piston area. Equation 3 shows why hydraulic actuation is suitable for the project. Increasing pressure or selecting a larger cylinder area increases the available drilling or impact force. In the final report, P should be replaced by the measured operating pressure and A should be calculated from the selected cylinder bore diameter.

The cylinder speed is related to hydraulic flow rate. For an ideal cylinder without leakage, the piston velocity can be approximated by

$$v = \frac{Q}{A} \tag{4}$$

where v is the piston velocity, Q is the volumetric flow rate into the cylinder chamber, and A is the effective piston area. This equation shows the tradeoff between force and speed. A larger piston area increases force for the same pressure, but it reduces speed for the same flow rate. A smaller piston area increases speed, but it reduces available force. The project therefore requires a balance between force output and the $5\ \mathrm{Hz}$ drill-bit motion target.

The pump does not directly determine the drill-bit direction. Instead, the pump provides continuous hydraulic pressure to the valve. The valve determines which side of the cylinder receives pressure and which side returns fluid to the reservoir. This is important because the pump can operate continuously while the valve switches the cylinder direction. Continuous pump operation avoids repeated motor startup events and allows the control system to focus on valve switching rather than motor switching.

The reservoir provides hydraulic fluid storage and allows returning fluid to circulate back to the pump inlet. Hoses and fittings connect the pump, valve, and cylinder. The hose routing is important for both performance and safety. Long or sharply bent hoses can increase pressure losses and can interfere with the vehicle-mounted mechanism. Loose hoses can also create hazards when the cylinder moves. Therefore, the hydraulic lines should be routed with enough slack for motion but should not be allowed to cross the drill-bit path or the cylinder linkage. Figure 6 shows the hydraulic reservoir used for fluid storage and return flow in the pump circuit.

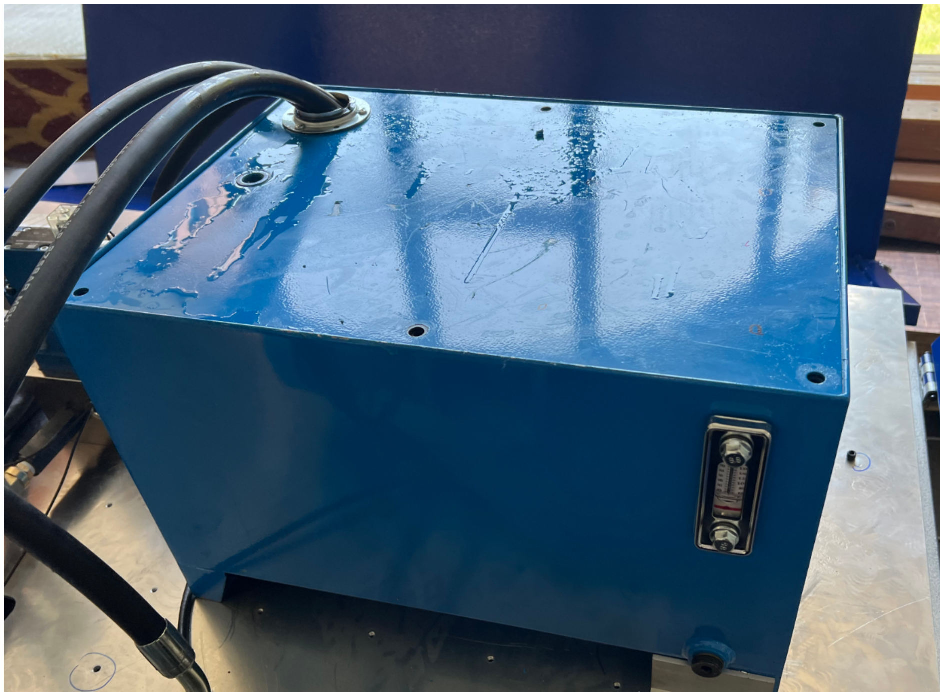

Figure 6: Hydraulic oil reservoir mounted on the vehicle platform. The reservoir stores the hydraulic fluid and provides the return path for the hydraulic circuit.

A pressure relief or pressure-control element should be included in the hydraulic circuit to prevent excessive pressure under blocked or high-load conditions. When the drill bit contacts the ground or another rigid surface, the cylinder may experience a rapid increase in load. Without pressure protection, the pump, hoses, valve, or cylinder could be exposed to pressure above their safe operating limits. The relief setting should be lower than the rated pressure of the weakest hydraulic component. In the report, this should be described using the actual component ratings and the measured operating pressure.

Figure **??** should show the final hydraulic circuit. The figure should include the reservoir, pump, relief valve or pressure-control component, directional valve, hydraulic cylinder, and return path. A simplified hydraulic schematic is more useful than a photograph in the Design section because it directly shows how fluid flows through the system. The full prototype photograph can still be used in the system overview or mechanical integration subsection.

The hydraulic supply subsystem is therefore responsible for providing the physical energy needed for drilling, while the control subsystem determines the timing and direction of that energy. This separation is a major advantage of the design. It allows the pump to be selected for pressure and flow, while the controller is selected for timing accuracy and valve-command generation.

2.4 Voltage-Based Valve Switching Control

The voltage-based valve switching control is the central control mechanism of the project. The hydraulic cylinder is not driven directly by an electric actuator. Instead, a microcontroller-and-PCB-based control circuit generates a fixed-frequency ± 10 V command signal for the servo proportional directional valve. The control circuit is powered by a 72 V to 5 V DC-DC converter, while the valve itself is powered by a separate 72 V to 24 V DC-DC converter. The valve converts the low-power command signal into a hydraulic flow-direction change.Figure 7 shows the servo proportional directional valve and its hydraulic connections in the assembled prototype.

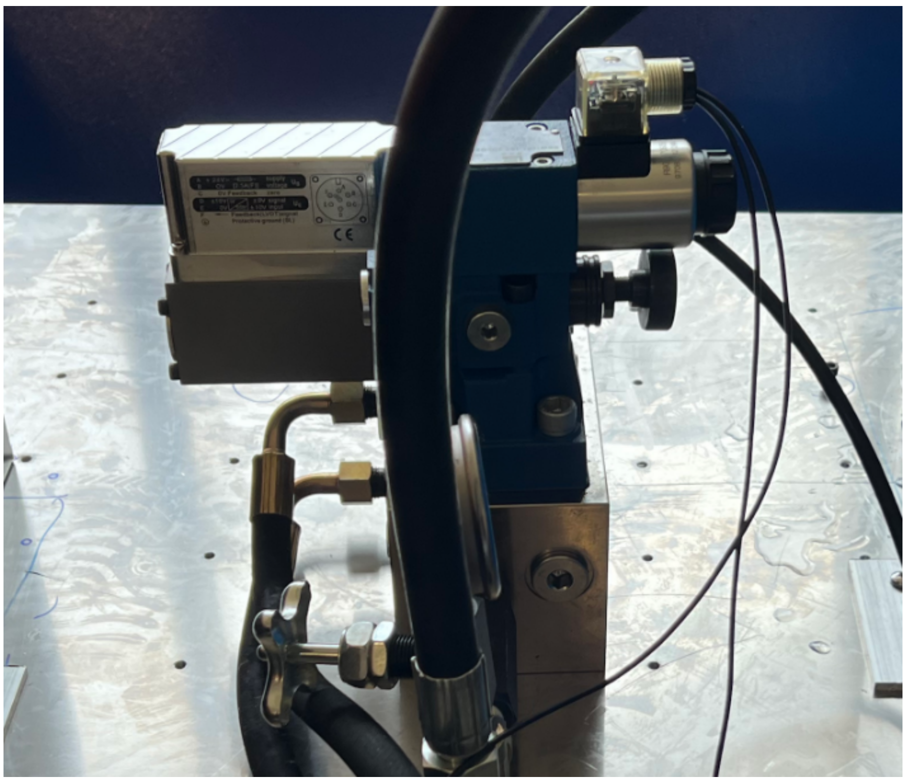

Figure 7: Servo proportional directional valve installed in the hydraulic circuit. The valve receives a 24 V power supply and a fixed-frequency ± 10 V command signal from the control circuit to change the hydraulic flow direction.

The control logic uses two alternating command states. In one state, the valve directs pressurized fluid to the cylinder chamber that produces the forward or downward drill-bit stroke. In the other state, the valve reverses the fluid path and drives the return stroke. These two states repeat periodically. The electrical command switches at 10 Hz, so the valve receives ten direction commands per second. However, one complete drill-bit reciprocation requires both command states, so the mechanical output frequency is 5 Hz, as shown previously in Equation 2.

In the implemented prototype, the control circuit is implemented with a microcontroller and PCB. The control electronics are powered from the regulated 5 V supply and generate a fixed-frequency ± 10 V command signal. This command signal is separate from the 24 V valve power supply. The 24 V supply powers the servo proportional valve, while the ± 10 V signal determines the commanded valve position and hydraulic flow direction.

A simple timing relation defines the control waveform. For a switching frequency of 10 Hz, the switching period is

$$T_{\text{switch}} = \frac{1}{f_{\text{switch}}} \tag{5}$$

Using $f_{\text{switch}} = 10$ Hz,

$$T_{\text{switch}} = \frac{1}{10 \text{ Hz}} = 0.1 \text{ s} \tag{6}$$

The valve command changes state every 0.1 s. A full drill-bit cycle requires two command states, so the drill-bit period is

$$T_{\text{drill}} = 2T_{\text{switch}} = 0.2 \text{ s} \tag{7}$$

and the corresponding drill-bit frequency is $1/T_{\text{drill}} = 5$ Hz. This time-domain explanation is useful in the report because it clearly distinguishes a direction-switching event from a complete mechanical cycle.

The control circuit should be verified with an oscilloscope before it is connected to the valve. This test should show the voltage command amplitude, frequency, duty cycle, and transition behavior. If the duty cycle is approximately 50%, the extension and retraction commands receive equal time. If the cylinder requires different times for extension and retraction because of unequal cylinder areas or load direction, the duty cycle can be adjusted in future versions. For the current prototype, a symmetric switching command provides a simple and repeatable initial control method.

The voltage switching control also provides a practical safety advantage because the control signal and the pump power path are separated. The control circuit only commands the valve and cannot create hydraulic motion without pump pressure. The main switch or air circuit breaker can disable the high-power path to the motor drive and pump, while the low-power control circuit remains a separate regulated subsystem. This layered structure reduces the risk that a low-power control fault alone could create unintended hydraulic motion.

Figure **??** should show the voltage switching logic. The figure should include the 72 V to 5 V converter, the microcontroller-and-PCB control circuit, the fixed-frequency ± 10 V output, the 72 V to 24 V valve power supply, the servo proportional directional valve, and the hydraulic cylinder. The figure should make clear that the 24 V line powers the valve, while the ± 10 V line is the valve command signal.

The main alternative to this approach would be a fully closed-loop servo-hydraulic system with position or force feedback. Such a system could provide more precise displacement or force control, but it would require a faster valve, additional sensors, a feedback controller, and more complex tuning. Another alternative would be an electric linear actuator, but achieving the required force and reciprocating speed would likely increase actuator size and cost. The selected voltage-switching hydraulic approach is simpler and better matched to the prototype goal: demonstrate a controllable vehicle-mounted hydraulic drilling mechanism within the available time and budget.

2.5 Hydraulic Cylinder and Drill-Bit Actuation

The hydraulic cylinder converts hydraulic pressure and flow into linear motion. In this project, the cylinder is the main actuator that drives the drill-bit or impact end. When the electro-hydraulic directional valve sends pressurized fluid to one side of the cylinder, the piston moves in one direction. When the valve reverses the flow, the piston moves in the opposite direction. This alternating extension and retraction creates the reciprocating motion required for the drill bit. Figure 8 shows the hydraulic cylinder mounted under the vehicle platform. This location allows the cylinder motion to be transferred toward the impact end while keeping the actuator supported by the vehicle frame.

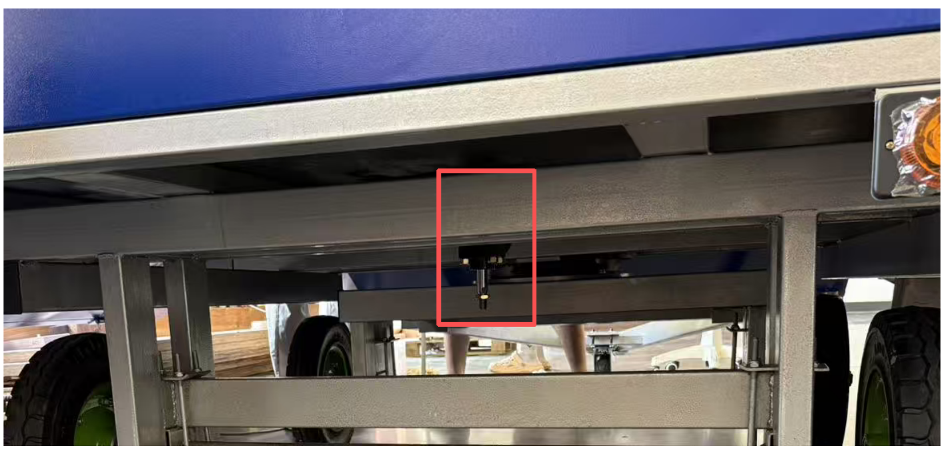

Figure 8: Hydraulic cylinder mounted under the vehicle platform. The cylinder is installed below the frame so that its linear motion can be transferred to the impact or drill-bit end while remaining mechanically supported by the vehicle structure.

The cylinder was selected because it can provide high linear force in a compact form. The available output force depends on the operating pressure and the effective piston area, as shown in Equation 3. The actual force differs between extension and retraction if the cylinder is a single-rod double-acting cylinder, because the rod reduces the effective area on one side of the piston. The extension force can be approximated as

$$F_{\text{extend}} = PA_{\text{bore}} \tag{8}$$

while the retraction force can be approximated as

$$F_{\text{retract}} = P(A_{\text{bore}} - A_{\text{rod}}) \tag{9}$$

where A_{bore} is the bore-side piston area and A_{rod} is the rod cross-sectional area. These equations should be evaluated using the selected cylinder dimensions and the measured hydraulic pressure. They also explain why the cylinder may not move identically in the two directions under the same pressure and flow conditions.

The drill-bit output frequency depends on both the valve command and the physical ability of the cylinder to complete the required stroke. The $10\ \text{Hz}$ valve command defines the intended timing, but the hydraulic cylinder must also have enough flow to move through the required displacement within each half-cycle. At $5\ \text{Hz}$, one full drill-bit cycle lasts $0.2\ \text{s}$, and each direction receives approximately $0.1\ \text{s}$ if the duty cycle is 50%. Therefore, the required average piston velocity for one stroke is approximately

$$v_{\text{req}} = \frac{L_{\text{stroke}}}{0.1\ \text{s}} \tag{10}$$

where L_{stroke} is the effective stroke length used during drilling. This value may be shorter than the full rated cylinder stroke if the mechanism only uses part of the cylinder travel. The required hydraulic flow can then be estimated from

$$Q_{\text{req}} = Av_{\text{req}} \tag{11}$$

Equations 10 and 11 connect the mechanical frequency target to the hydraulic pump and valve requirements. They show that a higher drill-bit frequency requires either a shorter stroke, a smaller cylinder area, or a higher flow rate. This tradeoff is one of the main design constraints of the system.

The drill-bit assembly must transfer cylinder motion to the contact point without excessive compliance or misalignment. If the connection between the cylinder and the drill bit is flexible, loose, or angled, part of the cylinder motion will be lost in deformation or side loading. Side loading is especially undesirable because hydraulic cylinders are designed primarily for axial loading. The mechanical linkage should therefore keep the drill-bit motion aligned with the cylinder axis as much as possible. The mounting bracket should also resist vibration caused by repeated $5\ \text{Hz}$ contact motion.

The drill-bit or impact end should be treated as the final output of the system rather than as a separate tool. Its shape, contact area, and stiffness affect the observed drilling or impact behavior. A sharper bit may concentrate force into a smaller area, while a broader impact head may distribute the force and reduce penetration. The final report should describe the selected end-effector geometry and explain why it was appropriate for the intended test surface.

The cylinder and drill-bit actuation subsystem should be verified by measuring or observing the output motion. The most direct verification method is high-speed video or frame-by-frame video analysis. The number of full extension-retraction cycles can be counted over a known time interval to confirm the $5\ \mathrm{Hz}$ output. If available, a limit switch, displacement sensor, accelerometer, or load cell can provide more quantitative verification. The minimum required evidence is that the drill bit completes repeated forward and return motion under hydraulic power at the intended operating condition.

2.6 Vehicle-Mounted Mechanical Integration

The final prototype was integrated on a vehicle-mounted platform rather than a stationary laboratory setup. This integration required the hydraulic, electrical, and mechanical subsystems to be fixed in stable positions so that the system could operate during movement and repeated impact actuation. The mounting structure supports the hydraulic cylinder, impact or drill-bit assembly, hoses, wiring, battery, motor drive, pump, reservoir, valve, and control electronics. During integration, the main design concerns were keeping the impact-end motion aligned with the intended contact direction, preventing hoses and wires from interfering with moving parts, and ensuring that the high-power electrical components and hydraulic components were securely mounted. As shown previously in Figure 2, the assembled prototype places the power, control, hydraulic supply, valve, and actuator modules on the vehicle frame to form a complete movable electro-hydraulic impact testing platform. This integration demonstrates that the project is not only a bench-top hydraulic circuit, but a functional vehicle-mounted prototype capable of converting electrical valve commands into repeated hydraulic impact-end motion.

3 Verification

3.1 Verification Overview

The verification process focused on three key functions of the prototype. First, the control circuit must generate the correct fixed-frequency $\pm 10\ \mathrm{V}$ command signal for the servo proportional directional valve. Second, the power system must provide the correct voltage levels: $72\ \mathrm{V}$ battery input, $24\ \mathrm{V}$ valve power, $5\ \mathrm{V}$ control-circuit power, and the inverter output for the brushless DC motor. Third, the complete system must convert the electrical valve command into repeated hydraulic cylinder and impact-end motion.

Because the system is based on voltage-controlled hydraulic switching, the most important verification target is the relationship between the $10\ \mathrm{Hz}$ electrical switching signal and the $5\ \mathrm{Hz}$ mechanical drill-bit output. The following tests were selected to verify this main design logic without adding unnecessary low-level measurements.

3.2 Valve Command Voltage Test

The first test verified the output of the voltage switching circuit. The valve-command output was measured before and during system operation to confirm that the control circuit could generate both positive and negative command voltages. The expected output was an alternating signal between $+10$ V and -10 V at 10 Hz.

This test is important because the electro-hydraulic directional valve changes the hydraulic flow direction according to the command voltage polarity. A positive voltage commands one valve direction, while a negative voltage commands the opposite valve direction. Therefore, this test verifies that the controller can command both extension and retraction of the hydraulic cylinder.

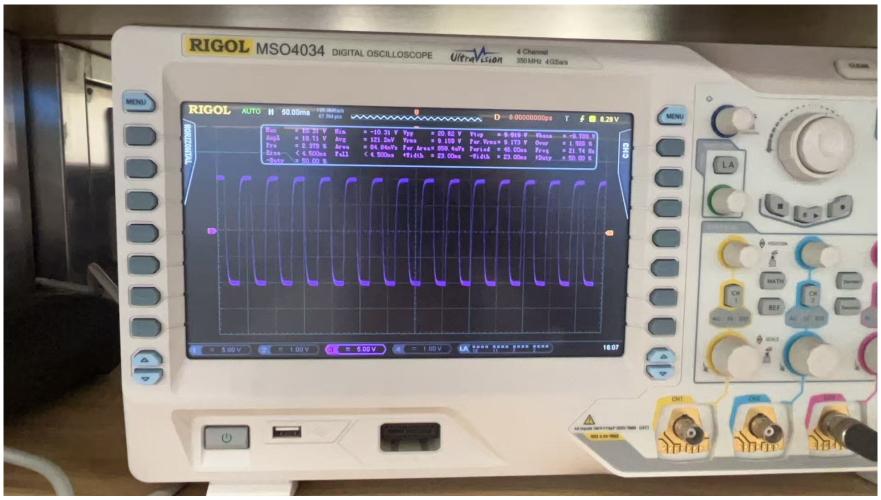

Figure 9: Oscilloscope measurement of the control output signal. The waveform shows an approximately ± 10 V square-wave command with a switching frequency of 10 Hz, satisfying the directional valve control requirement.

3.3 Power Supply Voltage Test

The second test verified the main power levels used by the system. The 72 V battery output was measured to confirm the input to the system. The output of the 72 V to 24 V DC-DC converter was measured to confirm that the servo proportional directional valve received its required valve power. The output of the 72 V to 5 V DC-DC converter was measured to confirm that the microcontroller and PCB control circuit received stable logic power. The DC inverter/motor drive output was also checked to confirm that the brushless DC motor received the required three-phase drive signal.

These measurements are important because the system depends on separate voltage levels for different functions. The motor drive supplies the hydraulic pump motor, the 24 V converter powers the servo proportional valve, and the 5 V converter powers the control circuit. If any of these voltage levels is incorrect, the system may fail to generate hydraulic pressure or may fail to command the valve correctly.

Table 2: Power supply voltage test

Measurement	Expected Value	Measured Value
Battery voltage	72 V nominal	78.0 V no load; 77.8 V during operation
Valve power supply	24 V DC	24.1 V
Control circuit supply	5 V DC	5.0 V

3.4 Integrated Motion Test

The final test verified the complete electrical-to-hydraulic-to-mechanical operation of the prototype. After the valve command and motor voltage were verified, the electro-hydraulic directional valve was connected to the hydraulic circuit. The system was then operated with the 10 Hz switching command.

The expected result was repeated extension and retraction of the hydraulic cylinder. Since one complete drill-bit cycle requires one extension stroke and one retraction stroke, the expected drill-bit frequency is half of the electrical switching frequency:

$$f_{\text{drill}} = \frac{f_{\text{switch}}}{2} \tag{12}$$

For the implemented prototype,

$$f_{\text{drill}} = \frac{10 \text{ Hz}}{2} = 5 \text{ Hz} \tag{13}$$

The drill-bit output frequency was measured from a recorded video by counting complete forward-return cycles over a known time interval. In the video measurement, the drill bit completed 21 full reciprocating cycles in 5.0 s. Therefore, the measured output frequency was

$$f_{\text{drill,measured}} = \frac{21 \text{ cycles}}{5.0 \text{ s}} = 4.2 \text{ Hz} \tag{14}$$

This value is lower than the ideal 5 Hz output expected from a 10 Hz electrical switching command. The difference is reasonable for the current prototype because the mechanical output frequency is affected not only by the command frequency but also by the hydraulic response time of the servo proportional directional valve, the available pump flow rate, fluid compressibility, hose pressure losses, cylinder friction, and the inertia of the drill-bit mechanism. In addition, the cylinder may not complete the full extension or retraction instantly after each voltage command transition, so the practical reciprocating frequency can be lower than the ideal value calculated from the electrical switching frequency.

Table 3: Integrated motion test

Item	Expected Result	Observed Result
Cylinder motion	Repeated extension and retraction	Verified by video observation
Electrical switching frequency	$10\ \mathrm{Hz}$	$10\ \mathrm{Hz}$
Drill-bit output frequency	$5\ \mathrm{Hz}$ ideal	$4.2\ \mathrm{Hz}$ measured

The measured result still verifies the main function of the system: the fixed-frequency $\pm 10\ \mathrm{V}$ valve command was successfully converted into repeated hydraulic cylinder motion and drill-bit reciprocation. However, the result also shows that the hydraulic and mechanical response limits of the prototype should be considered when comparing the theoretical command frequency with the actual mechanical output frequency.

These three tests verify the main design claim of the project. The microcontroller-and-PCB-based control circuit generates the required fixed-frequency $+10\ \mathrm{V}$ and $-10\ \mathrm{V}$ valve commands. The power system provides the required $72\ \mathrm{V}$, $24\ \mathrm{V}$, and $5\ \mathrm{V}$ levels, as well as the inverter output needed to drive the brushless DC motor and hydraulic pump. The integrated system then converts the electrical valve command into repeated hydraulic cylinder and impact-end motion.

4 Cost Analysis

Table 4 summarizes the material cost of the Movable Impact Testing Platform. All listed components had been purchased by the time the final prototype was assembled on the vehicle. The cost includes the hydraulic, electrical, sensing, control, and mechanical integration components used to build the prototype.

Table 4: Material cost analysis of the Movable Impact Testing Platform

Item	**Qty.**	**Unit Cost (CNY)**	**Total (CNY)**	**Status**
Hydraulic oil tank	1	187.10	187.10	Purchased
Hydraulic pump	1	410.00	410.00	Purchased
Load cell sensor	1	198.00	198.00	Purchased
Motor controller / DC inverter	1	946.51	946.51	Purchased
Motor	1	1970.00	1970.00	Purchased
Battery pack	1	5998.05	5998.05	Purchased
72 V to 5 V DC-DC converter	1	75.00	75.00	Purchased
72 V to 24 V DC-DC converter	1	144.00	144.00	Purchased
Proportional directional valve	1	7100.00	7100.00	Purchased
Cooling system	1	600.00	600.00	Purchased
Relief valve	1	1200.00	1200.00	Purchased
Valve block, non-standard custom	1	2100.00	2100.00	Purchased
Hydraulic cylinder	1	350.00	350.00	Purchased
PCB board and control components	1	150.00	150.00	Purchased
Hydraulic hoses and fittings	1 set	300.00	300.00	Purchased
Base plate and support frame materials	1 set	600.00	600.00	Purchased
Wheels and locking mechanism	1 set	250.00	250.00	Purchased
Fasteners, wiring, and miscellaneous parts	1 set	200.00	200.00	Purchased
		Total Material Cost	**22778.66**	

The total material cost of the final prototype was 22778.66 CNY. The largest cost items were the battery pack, proportional directional valve, motor, valve block, relief valve, and motor controller. These components dominate the total cost because the platform requires a high-power electrical supply, hydraulic pressure generation, and controlled valve actuation. Smaller items, including the DC-DC converters, PCB board, hoses, fasteners, wiring, and support materials, were also necessary for system integration but contributed less to the overall cost.

This cost estimate represents the prototype-level material cost. A production version could reduce the unit cost through bulk purchasing, simplified mechanical packaging, and integration of the electrical and hydraulic subsystems into a more compact enclosure.

In addition to the material cost, labor cost was estimated using the ECE 445 guideline formula:

$$\text{Labor Cost} = \text{Hourly Rate} \times \text{Hours Worked} \times 2.5 \tag{15}$$

The hourly rate was assumed to be $20\ \mathrm{USD/h}$, and each team member was estimated to have spent approximately $110\ \mathrm{h}$ on the project. The factor of 2.5 accounts for overhead costs associated with professional engineering labor. This labor cost is not an actual project expense, but an estimate of the engineering cost required to develop the prototype in a professional setting.

Table 5: Estimated labor cost

Team Member	**Hourly Rate (USD/h)**	**Hours Worked**	**Labor Cost (USD)**
Shangyu Wang	20	110	5500
Bingkun Fu	20	110	5500
Feiyu Tang	20	110	5500
Yihang Shen	20	110	5500
		Total Estimated Labor Cost	**22000**

5 Conclusion

This project developed the Movable Impact Testing Platform, a vehicle-mounted electro-hydraulic prototype that converts an electrical valve command into repeated hydraulic impact-end motion. The final prototype uses a $72\ \mathrm{V}$ battery, a main switch or air circuit breaker, and a DC inverter/motor drive to power the brushless DC motor and hydraulic pump. A separate $72\ \mathrm{V}$ to $24\ \mathrm{V}$ DC-DC converter powers the servo proportional directional valve, while a $72\ \mathrm{V}$ to $5\ \mathrm{V}$ DC-DC converter powers the microcontroller-and-PCB-based control circuit. The control circuit generates a fixed-frequency $\pm 10\ \mathrm{V}$ command signal for the valve, and the valve changes the hydraulic flow direction to produce repeated cylinder motion.

The main accomplishment of the project is the successful integration of the electrical, hydraulic, and mechanical subsystems into one working prototype. The power subsystem supplies the motor drive, servo valve, and control circuit through separate voltage paths. The control circuit generates the directional valve command, the hydraulic circuit converts pump output into cylinder motion, and the mechanical structure transfers this motion to the impact or drill-bit end. The verification tests focus on the most important system functions: the $\pm 10\ \mathrm{V}$ valve-command output, the required system voltage levels, and the integrated reciprocating motion.

Several limitations remain in the current prototype. The system demonstrates controlled hydraulic reciprocation, but more detailed force, pressure, temperature, and long-duration reliability tests would be needed before the design could be considered ready for field deployment. Future work should improve the mechanical mounting structure, add stronger protection for the control electronics, organize the hydraulic hoses and wiring into a safer layout, and use additional sensors to measure cylinder displacement or drill-bit force during operation. A closed-loop control system could also be added if more precise motion or force control is required.

Ethical and safety considerations are important because the prototype combines high electrical power, hydraulic pressure, and moving mechanical parts. The design should be operated only with secure wiring, properly rated hydraulic components, and a clear safety area around the drill-bit mechanism. These precautions are consistent with the IEEE Code of Ethics, which emphasizes public safety, responsible engineering practice, and honest reporting of system capabilities and limitations **ieee˙code˙ethics**. The broader impact of the project is that a compact vehicle-mounted drilling or impact mechanism could reduce the need for larger stationary testing equipment, but it must be designed carefully to avoid electrical, hydraulic, and mechanical hazards.

Overall, the prototype demonstrates the feasibility of using voltage-based valve switching to control a mobile hydraulic drilling mechanism. The final system shows that a $10\ \mathrm{Hz}$ electrical switching command can be translated into an approximately $5\ \mathrm{Hz}$ drill-bit reciprocating output through an electro-hydraulic directional valve and hydraulic cylinder.

References

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