Engine Simulation and Diagnostic Test Bed

Final Design Report

Sponsor: Jeff Lehmkuhl

Date: 12/6/2011

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Statement of Disclaimer

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List of Nomenclature

**ASM:** Acceleration Simulation Mode smog testing. ASM smog testing measures CO, HC (hydrocarbon), and NOx emissions in the exhaust gas.

**Bellhousing:** Bell-shaped part of the transmission which covers the flywheel and bolts to the engine block

**Coupler:** A mechanical connection between two shafts, allowing power transmission while allowing for some misalignment

**Dynamometer:** A dynamometer is a device for measuring the power output of an engine.

**NIOSH:** National Institute for Occupational Safety and Health. NIOSH is a government agency responsible for conducting research and making recommendations to prevent work-related injuries and illness.

**OBD-I:** On-Board Diagnostics, generation one interface

**OSHA:** Occupational Safety & Health Administration. OSHA is a government agency that creates and enforces safety and health standards in the workplace.

**PCM:** Powertrain Control Module. The PCM is an electronic control unit that is responsible for the functions of the engine and transmission.

**QFD:** Quality Function Deployment, a criteria selection process

**Spider:** An insert which transmits power between two halves of a coupler while permitting misalignment compliance.
Chapter 1 - Introduction

1.1 Abstract

The San Luis Obispo High School Automotive Technology Shop has a 1991 Buick V-6 Engine which uses the OBD-I (On-Board Diagnostics, generation one) standard for diagnostic testing and troubleshooting. In order to expand the capabilities of the Automotive Technology program and allow students to gain first-hand experience troubleshooting older OBD-I engines, the shop has a need for this engine to become part of a diagnostic and troubleshooting test bed. Some of the diagnostic tests the students will perform include checking cylinder pressure, monitoring coolant system temperatures, interpreting error codes from the OBD-I system, and conducting enhanced smog testing. All of these tests can be performed without modification to the engine except for the enhanced smog testing, which requires the addition of a separate system to place a load on the engine. The project was proposed by Jeff Lehmkuhl, the San Luis Obispo High School Automotive Shop Instructor. The project was primarily funded by the San Luis Obispo High School Automotive Technology Shop and the Mechanical Engineering Department at Cal Poly. Additional individual monetary and material donations also helped make the project possible. A list of supporting sponsors can be found in Appendix A. The students who use the shop will ultimately benefit from the additional training and experience this tool will offer as part of the Automotive Technology Program. The goal of the project is to mount this engine in a self-contained, movable test bed to facilitate diagnostic testing and troubleshooting of real-world problems on an OBD-I based system.

1.2 Problem Definition

Our goal in this project is to create a product that is useful for training Automotive Technology students in diagnostic and troubleshooting procedures on older OBD-I based engines. The system will be built around the 1991 Buick 3800 V-6 engine and transaxle from the High School Automotive Shop. The final system will consist of a cart containing all the necessary subsystem components (fuel cell, radiator with fan, exhaust and battery) in order to safely operate the engine. The engine will be mounted at a raised height compared to the stock vehicle, so that it is easier to reach and work on. The cart will have a control interface with relevant gauges (tachometer, volts, oil pressure, and coolant temp) and controls (throttle, ignition switch, kill switch) to run, monitor, and read feedback from the engine. Near the control interface will be a work surface for holding tools and manuals during use. In order to produce a system capable of being used with little instruction, tutorial guides will be created to allow students to independently learn common troubleshooting procedures.
1.3 Specification Development

In order to measure the success of our final product, engineering specifications have been developed based upon customer requirements. These specifications are created as measurable, verifiable criteria that should be met to create a successful product. A Quality Function Deployment (QFD) method analysis was conducted to define and prioritize customer requirements, compare the performance of currently available competitive products, and develop verifiable engineering specifications related to customer requirements. The QFD chart can be seen in Appendix A.

Table 1 below shows the proposed specifications resulting from the QFD analysis. In addition, the specification requirement is listed, its risk to the success of the project is subjectively rated, and proposed methods of compliance verification are listed. There are four main types of compliance verification. Analysis (A) would consist of design calculations verifying performance. Testing (T) will be developed and conducted for some specifications. Similarity (S) can be used to check our design against similar products and components available. Also Inspection (I) can be conducted to measure and verify specifications.
Table 1. Proposed Project Engineering Requirements

<table>
<thead>
<tr>
<th>Spec. Number</th>
<th>Parameter Description</th>
<th>Requirement or Target (units)</th>
<th>Tolerance</th>
<th>Risk</th>
<th>Compliance</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Frame Strength, load of engine and transaxle</td>
<td>1000 lb</td>
<td>Min.</td>
<td>H</td>
<td>A,T,S</td>
</tr>
<tr>
<td>2</td>
<td>User Interface contains Oil Pressure, Coolant Temp, Ignition-On Light, Tachometer, Check Engine Light, OBD DLC, Ignition Switch</td>
<td>Yes/No</td>
<td>N/A</td>
<td>M</td>
<td>I</td>
</tr>
<tr>
<td>3</td>
<td>User Interface Includes a Work Surface Between 36&quot; and 46&quot; off the ground, at least 20&quot; wide and 10&quot; deep.</td>
<td>Yes/No</td>
<td>N/A</td>
<td>L</td>
<td>A,I</td>
</tr>
<tr>
<td>4</td>
<td>Engine will be mounted at a raised height for ease of access for work. Height of Spark Plugs will be at least 30&quot; above the ground.</td>
<td>Yes/No</td>
<td>N/A</td>
<td>M</td>
<td>A,I</td>
</tr>
<tr>
<td>5</td>
<td>System Capable of Being Maneuvered by a Single Average Person</td>
<td>Yes/No</td>
<td>N/A</td>
<td>L</td>
<td>A,T</td>
</tr>
<tr>
<td>6</td>
<td>Engine Includes Key Switch to Prevent Unauthorized Operation</td>
<td>Yes/No</td>
<td>N/A</td>
<td>M</td>
<td>I,T</td>
</tr>
<tr>
<td>7</td>
<td>Moving Components are effectively caged to prevent injury. Check against OSHA Spec.</td>
<td>Yes/No</td>
<td>N/A</td>
<td>H</td>
<td>I,T</td>
</tr>
<tr>
<td>8</td>
<td>No Electrical or Mechanical Modifications are Made of the Stock Engine or Engine Wiring Harness.</td>
<td>Yes/No</td>
<td>N/A</td>
<td>L</td>
<td>I</td>
</tr>
<tr>
<td>9</td>
<td>Additional Raw Material Costs.</td>
<td>$500</td>
<td>Max.</td>
<td>M</td>
<td>A</td>
</tr>
<tr>
<td>10</td>
<td>Additional Components Cost.</td>
<td>$500</td>
<td>Max.</td>
<td>M</td>
<td>A</td>
</tr>
<tr>
<td>11</td>
<td>Final System will Include Step-By-Step Instructional Tutorials.</td>
<td>4 Tutorials</td>
<td>Min.</td>
<td>L</td>
<td>I,T</td>
</tr>
<tr>
<td>12</td>
<td>System will have all necessary subsystems mounted on-board (Ex. Fuel, Exhaust, Cooling, Battery, Etc.) for Standalone Operation</td>
<td>Yes/No</td>
<td>N/A</td>
<td>M</td>
<td>I,S,T</td>
</tr>
<tr>
<td>13</td>
<td>Entire System will not exceed 4' wide x 6.5' long x 5' tall</td>
<td>Yes/No</td>
<td>N/A</td>
<td>L</td>
<td>A,I</td>
</tr>
<tr>
<td>14</td>
<td>System includes loading equipment to simulate engine load at 15 and 25 mph.</td>
<td>Yes/No</td>
<td>N/A</td>
<td>H</td>
<td>A,I,T</td>
</tr>
</tbody>
</table>
The engine, cradle, and transaxle with transmission fluid weigh approximately 750 lbs. Based on this measurement and our customer requirement that the test stand should support the engine and all components, we chose a preliminary minimum strength requirement of 1000 lbs for Specification Number 1 in the table above. Based on our QFD analysis, the requirement that the test stand should be an effective and independent learning tool ranked highest among the needs of both Mr. Lehmkuhl and the students. The gauges and work surfaces need to be an appropriate height from the ground and of sufficient size to hold tools and tutorials while the students are using the test stand. To satisfy these requirements, we determined that the test stand should have a work surface with the dimensions outlined in Specification Number 3. The heights specified for the work surface and the engine spark plugs in Specifications Number 3 & 4 were determined using guidelines from NIOSH. Safety is a primary requirement for the project, especially since teenagers will be using the test stand. To help ensure that the test stand will be safe to operate, we decided it would be prudent to incorporate a Lock Out-Tag Out system as addressed in Specification Number 6. Specification Number 7 requires that any exposed moving parts, such as pulleys, will need to be covered.

The proposed specifications, agreed upon with the customer, will not be modified without further analysis and discussion. The specifications will be used in developing test plans and performing final testing to the product to verify its performance.

1.4 Project Management

Table 2 outlines some of the anticipated management roles necessary during the project progress. We will all be contributing throughout the process, however assigning individual responsibilities helps maintain on-schedule progress through individual accountability. It will be up to the individual in charge of an area to track progress and assign work to others in the group to make sure that area remains on schedule.

<table>
<thead>
<tr>
<th></th>
<th>Jeff Lewis</th>
<th>Cody Scott</th>
<th>Todd Smith</th>
</tr>
</thead>
<tbody>
<tr>
<td>Main point of contact with sponsor</td>
<td>Manage CAD model</td>
<td>Create and implement a testing plan</td>
<td></td>
</tr>
<tr>
<td>Manage team meetings</td>
<td>Oversee manufacturing</td>
<td>Design electrical system/wiring diagram</td>
<td></td>
</tr>
<tr>
<td>Analysis, sizing of braking system components</td>
<td>Manage any outsourced manufacturing processes</td>
<td>Budget/accounting</td>
<td></td>
</tr>
<tr>
<td>Braking sys. detailed drawings</td>
<td>Frame detailed drawings</td>
<td>Testing plan and tutorial development</td>
<td></td>
</tr>
<tr>
<td>Braking system manufacturing</td>
<td>Frame manufacturing</td>
<td>UI layout</td>
<td></td>
</tr>
<tr>
<td>Source additional materials</td>
<td>System Integration - Final Assembly</td>
<td>UI detailed drawings</td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td>UI assembly, wiring</td>
<td></td>
</tr>
</tbody>
</table>
Table 3 below shows important milestones in the project development process. Each of the Reports will be a formal document sent to both the sponsor and the project advisor. Meetings listed are to review the relevant reports or project progress and receive feedback from the sponsor pertaining to the project progress.
<table>
<thead>
<tr>
<th>Quarter</th>
<th>Week Of</th>
<th>Type</th>
<th>Milestone Description</th>
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</thead>
<tbody>
<tr>
<td>1</td>
<td>1/31/2011</td>
<td>Report</td>
<td>Project Proposal / Requirements Document</td>
</tr>
<tr>
<td></td>
<td>3/4/2011</td>
<td>Design</td>
<td>Select Braking System</td>
</tr>
<tr>
<td></td>
<td>3/14/2011</td>
<td>Meeting</td>
<td>Conceptual Design Review</td>
</tr>
<tr>
<td></td>
<td>3/29/2011</td>
<td>Milestone</td>
<td>Finalize Engine Subsystem Design</td>
</tr>
<tr>
<td></td>
<td>4/8/2011</td>
<td>Milestone</td>
<td>Finalize Frame Design</td>
</tr>
<tr>
<td></td>
<td>4/15/2011</td>
<td>Milestone</td>
<td>Finalize Braking System Design</td>
</tr>
<tr>
<td></td>
<td>3/30/2011</td>
<td>Task</td>
<td>Order Needed Parts and Materials</td>
</tr>
<tr>
<td></td>
<td>4/22/2011</td>
<td>Meeting</td>
<td>Critical Design Review</td>
</tr>
<tr>
<td></td>
<td>4/22/2011</td>
<td>Milestone</td>
<td>Start Build Phase</td>
</tr>
<tr>
<td></td>
<td>4/28/2011</td>
<td>Milestone</td>
<td>Finish Frame Manufacturing</td>
</tr>
<tr>
<td></td>
<td>5/12/2011</td>
<td>Milestone</td>
<td>Finish Braking System Manufacturing</td>
</tr>
<tr>
<td></td>
<td>5/9/2011</td>
<td>Milestone</td>
<td>Complete Mounting of Engine to Frame</td>
</tr>
<tr>
<td></td>
<td>5/20/2011</td>
<td>Milestone</td>
<td>Complete Engine Subsystem Installation</td>
</tr>
<tr>
<td></td>
<td>5/12/2011</td>
<td>Milestone</td>
<td>Complete Final Assembly</td>
</tr>
<tr>
<td>2</td>
<td>9/19/2011</td>
<td>Report/Meeting</td>
<td>Present Test Plan to Sponsor</td>
</tr>
<tr>
<td></td>
<td>9/16/2011</td>
<td>Meeting</td>
<td>Discuss Tests for Tutorial Development</td>
</tr>
<tr>
<td></td>
<td>9/24/2011</td>
<td>Testing</td>
<td>Frame Load Capacity Testing</td>
</tr>
<tr>
<td></td>
<td>10/8/2011</td>
<td>Testing</td>
<td>Ergonomic and Safety Testing</td>
</tr>
<tr>
<td></td>
<td>12/1/2011</td>
<td>Presentation</td>
<td>Senior Project Design Expo</td>
</tr>
</tbody>
</table>
Engine test stands are typically used on a small scale by individual automotive enthusiasts to test an engine before its installed into a vehicle. They provide all the necessary fuel, cooling, and electrical sub-systems to start and run the engine for a limited period of time. Engine test stands consist of a support structure to hold the engine and all the associated sub-systems in place and they provide a platform for testing applications. The students in the SLO High School Automotive Technology program learn how to diagnose and fix common engine problems, so the engine test bed will be tailored to that purpose, including connections for an OBD-I scanner and a method of loading the engine to perform enhanced smog testing.

The Engine Simulation and Diagnostic Test bed for the SLO High School Automotive Technology Shop is to be designed around a GM 3.8L Series I 3800 V6 engine. The engine was transversely mounted in a 1991 front wheel drive Buick Park Avenue and includes the trans-axle and engine cradle. From 1991 – 1996, the base model Park

Figure 1. Diagram of basic components in an Engine Test Stand

Note: Transmission is not required on engine test stands. Where included, loading system may be connected to transmission output.
Avenue came stock with the 170 horsepower 3.8L Series I 3800 while a supercharged version was offered from 1992 and on. The Series I 3800 was a very popular engine used in 14 other cars, including models from Chevrolet, Pontiac, and Oldsmobile.

2.2 Existing Products

Several products currently exist which fulfill a similar need to that of our customer. Two examples, the Easy-Run Professional Series Engine Test Stand and the Larin Mobile Engine Testing Stand, were chosen to use as a comparative benchmark for our future design. Both of these engine test stands provide a mobile platform to mount an engine for testing applications. The Easy-Run Engine Test Stand includes a radiator, 3 gallon fuel cell, remote oil filter, battery tray, and rolls on polyurethane casters. An adapter can be purchased separately to accommodate a longitudinally mounted automatic transmission as well. The Larin Mobile Engine Testing Stand does not include accommodations for a transmission or a radiator, however it does have an adapter to attach a standard garden hose to the engine's cooling system in lieu of a radiator. It also has a 1 gallon fuel tank and rolls on steel casters. These test stands are primarily intended for running older carbureted engines, and lack accommodations for the additional components and electronic interfaces in modern computer-controlled engines.

Figure 2. Easy-Run Engine Test Stand (www.easy-run.com)

Figure 3. Larin Mobile Engine Testing Stand (www.mygaragestore.com/detail.aspx?ID=1011)
2.3 Engine Loading Systems for Smog Testing Requirement

The SLO High School Automotive Technology Shop would like to perform Acceleration Simulation Mode (ASM) smog testing with the Engine Simulation and Diagnostic Test Stand. ASM smog testing puts a load on the engine at 15 mph and 25 mph while measuring CO, HC (hydrocarbon), and NO\textsubscript{x} emissions in the exhaust gas.\textsuperscript{[3]} Based on data provided from the Cal Poly Ground Vehicles course in Figure 4 below, an average sedan expends approximately 12 horsepower to maintain a speed of 25 mph against rolling resistance and aerodynamic drag. To simulate these conditions, smog testing facilities use an inertial dynamometer to load the engine during ASM tests. Inertial dynamometers consist of fixed inertial weights inside of rollers that provide resistance to the car’s drive wheels. Sometimes an electric motor is coupled to the rollers to provide additional loading specific to the vehicle being tested. Several other options exist for providing load to an engine including hydraulic brakes, water brakes, fan brakes, electric motors, and caliper-disc brakes.

![Figure 4. Required Engine Power, Including Rolling Resistance and Aerodynamic Drag](image)

A hydraulic braking system uses a hydraulic pump connected to the engine along with a valve to restrict flow, providing a load. Water brakes consist of a rotor connected to the engine output shaft and enclosed in a housing. Water is pumped through the housing and dissipates power from the engine. A fan brake uses a large fan connected to the engine to generate resistance via airflow. AC and DC electric motors can be used to provide a load if they are connected to the engine and are operated as generators, converting the engine’s rotational power to electrical power. Motor control units adjust the load the motor provides to the engine.\textsuperscript{[4]} Caliper-disc brakes can also be used to provide resistance by simply applying the brakes while the engine is running.
Commercial dynamometer systems can be purchased in a variety of sizes and types for different applications, from go-karts to speed boats. The DYNOmite Kart "Lite" Kit, sold by Land and Sea, is a water brake dynamometer for use with small go-karts. It comes with the water brake absorber, a strain gage-equipped torque arm, a data acquisition computer with software. It can handle loads ranging from 0-30 horsepower. Figure 5 shows an example DYNOmite water brake system.

![Figure 5. DYNOmite Water Brake Dynamometer System](http://www.land-and-sea.com/images/dyno/diy/sample_water-brake_absorber_plumbing_600.jpg)

2.4 Specific Technical Data

Our engineering specifications were derived from a combination of measurements collected from the engine and cradle, and background research on testing methods and similar systems. In addition, we spoke with Mr. Lehmkuhl to develop a list of features and requirements to incorporate into our diagnostic test stand in order to meet his needs as well as the needs of the students. The weight of the engine, transaxle, and cradle, with transmission fluid, but without the engine coolant and oil was approximately 750 lbs. We initially measured the rough outside dimensions of the engine and cradle at 35"l x 24"w x 26"h. We are considering removing the engine from the transaxle to simplify the final system. The engine alone has total dimensions of 23.0"l x 24.5"w x 25.5"t. The complicated external geometry of the engine makes it difficult to gather specific dimensions. In order to design around the geometry, we have utilized scaled images to create a complete SolidWorks model of the relevant external features and geometry. The engine model is shown in Figure 6, below.
2.5 Applicable Safety Standards

A necessary requirement of the project is the safe operation of the engine diagnostic test stand. Relevant risks that must be considered include rotating machinery, hot exhaust components, high-pressure hydraulic fluid, flammable fuel storage and delivery, onboard chemical battery risks, and proper venting of exhaust gasses.

The Occupational Health and Safety Administration (OSHA) has set forth occupational safety guidelines for mechanical power transmission machinery in OSHA 29 CFR 1910 Subpart O. Subpart O contains detailed guidelines for safeguarding rotating machine elements such as pulleys, chain drives, and flywheels. The engine should be as accessible as possible on the test stand, but safety precautions will need to be considered in the design to prevent personal injury to the operator and anyone nearby from belt drives and other exposed moving components on the engine.

Subpart J of OSHA 29 CFR 1910 also contains relevant guidelines for controlling hazardous energy in electrical and mechanical systems with lockout/tagout devices. The test stand will need to have a lockout system, such as an ignition lock with a key, to prevent unauthorized use.

In addition to mechanical safety considerations, the test stand should be designed with appropriate ergonomics in mind. The National Institute for Occupational Safety and Health (NIOSH) has created guidelines, for standard work surface heights, based on application, to help prevent work related injuries.
Chapter 3 - Design Development

3.1 Preliminary Frame Design

Based on our initial measurements, we have created a preliminary design for the frame and work surfaces of the test stand. The design of the test stand is subject to change as the design of the loading system moves forward since they are both dependent upon each other.

We used a fairly subjective but straightforward approach to selecting a material for the structure of the test stand. Our design specifications state that the test stand must be able to support at least 1000 lbs. and the cost of materials should not exceed $500. Based on these specifications, along with the fact that weight is not a critical factor, and that the San Luis Obispo High School has a supply of steel tubing readily available to us, we chose steel. Steel is easily capable of supporting 1000 lbs. and is relatively inexpensive compared to aluminum or composite materials if we need to buy additional tubing. In addition, steel is durable, easy to work with, and provides a lot of options for applying a surface finish.

The preliminary frame design focuses on meeting the relevant engineering specifications for accessibility, size, and strength. The frame includes a work surface and space to incorporate a user interface. It also includes a place to mount the radiator in front of the engine. Castors will be at each of the four corners for maneuverability. The main frame rail is raised up 12” above the top of the castors to allow for easier access underneath the engine. The engine will be mounted by two brackets, one using existing engine mounts on the accessory side of the engine, while the other uses the bellhousing bolt pattern on the engine block. Figure 7 shows the current design of the steel frame without castors or brackets for the engine and components. Figure 8 shows the engine and radiator positioned in the frame.
The frame will be constructed using 2” x 2” x .120” square steel tubing for the main structure and supports because of its strength and availability.
We have developed 3 top concepts for our engine loading system:

- Disc brake/caliper loading system
- Single hydraulic pump loading system
- Dual hydraulic pump loading system

The disc brake/caliper loading system consists of mounting large disc brake rotors and calipers to the stock front suspension and connecting the CV joints to the transmission outputs. The calipers would be connected to a master cylinder with a lever that could be used to engage the calipers and apply a braking force to the engine. The disc brake/caliper loading system would be easy to implement since it would use much of the stock suspension and we wouldn’t have to worry about alignment issues. A pressure gage can be installed at the caliper bleeder valve to set the pressure required to apply the braking force necessary to dissipate 8-12 horsepower. The main concerns with this system are the amount of heat generated in the rotors over the duration of the smog test, safety factors, and the frequency of brake pad replacement. We performed an initial thermal analysis of the disc brake/caliper system with a summary of our results in Table 4 on the next page. The equations used to determine the heat transfer to the disc brakes are in the appendix.
Table 4: Results of Thermal Analysis for a Single Brake Disc Rotor

<p>| | |</p>
<table>
<thead>
<tr>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>Room Air Temp (°F)</td>
<td>60.00</td>
</tr>
<tr>
<td>Braking Power (HP)</td>
<td>6.00</td>
</tr>
<tr>
<td>Braking Power (BTU/s)</td>
<td>4.24</td>
</tr>
<tr>
<td>Rotor Mass (lbm)</td>
<td>10.00</td>
</tr>
<tr>
<td>Rotor cp (Btu/lbm °F)</td>
<td>0.11</td>
</tr>
<tr>
<td>Rotor Diameter (in)</td>
<td>13.00</td>
</tr>
<tr>
<td>Rotor Surface Area (in²)</td>
<td>265.50</td>
</tr>
<tr>
<td>Ventilation Factor fv</td>
<td>-</td>
</tr>
<tr>
<td>Temp @ 1 min Cont. Op. (°F)</td>
<td>291.00</td>
</tr>
<tr>
<td>Temp @ 5 min Cont. Op.(°F)</td>
<td>897.80</td>
</tr>
<tr>
<td>Temp @ 8 min Cont. Op.(°F)</td>
<td>999.00</td>
</tr>
</tbody>
</table>

The single hydraulic pump loading system uses a single gear pump attached to the accessory side of the engine. The pump shaft is fitted with a pulley and provides resistance to the engine using the serpentine belt which is normally used to drive the alternator, A/C, and other accessories. High pressure hydraulic tubing supplies the pump with hydraulic fluid from a reservoir. Heat transferred to the hydraulic fluid is dissipated through a heat exchanger on the low pressure side of the system before re-entering the reservoir. The level of resistance to the engine is controlled using a flow regulating valve and the power dissipated can be calculated from pressure gage and flow-meter readings. The single hydraulic pump loading system is simple, compact, and requires relatively low maintenance. Our main concerns with this design are safety issues regarding the serpentine belt and the heat generated in the pump and the heat exchanger. Figure 11 shows the components in a hydraulic braking system, optionally showing both one and two pump configurations.

![Figure 11. Schematic of Hydraulic System for Engine Loading](image-url)
The dual hydraulic pump loading system uses two gear pumps, each coupled to one of the transmission outputs. This design is very similar to the single gear pump system in that fluid is delivered to the pumps from a reservoir through high pressure tubing and controlled using a flow regulating valve. The pumps will most likely be bolted to a custom bracket which is attached to the frame of the test stand and coupled to the transmission outputs with custom machined shaft couplings. The dual hydraulic pump loading system should require little to no maintenance and be able to dissipate more power than the single hydraulic pump system.

A gear pump is ideal for the hydraulic system since it is a fixed-displacement pump. The fluid pumped will be directly proportional to the speed of the input, and the pressure can be regulated by a needle valve. The input work into the pump can be calculated based on the pressure, flow rate, and efficiency of the pump. Table 5 on the next page gives a summary of the horsepower developed in the Prince SP20 gear pump at a specified pressure and speed. The full set of calculations to determine the horsepower of the pump are in the appendix.

Table 5: Gear Pump Horsepower Calculation Results

<table>
<thead>
<tr>
<th>Prince SP20 Gear Pump</th>
</tr>
</thead>
<tbody>
<tr>
<td>Displacement (cu. In/rev)</td>
</tr>
<tr>
<td>Pressure (psi)</td>
</tr>
<tr>
<td>Speed (rpm)</td>
</tr>
<tr>
<td>Horsepower</td>
</tr>
</tbody>
</table>

This shows a similar pump would be capable of loading the engine adequately for the required test in the one pump system, and a smaller pump could even be used in the two pump system.

3.3 Loading System Selection

As shown in Table 6 below, we were able to narrow down our top concepts to two choices. Within the matrix we evaluated the safety, cost, maintenance, simplicity of operation, simplicity of design, simplicity of manufacturing, and likelihood of failure of each loading system by assigning a rank from 0 to 5, 0 being poor and 5 being excellent. Each category was also given a weighting factor between 1 and 5 based upon how important we thought it was to the overall design. Once each system was ranked, their scores were totaled and given a percentage based off a perfect score. The disc brake/caliper loading system scored the lowest with 58% while the dual pump and single pump loading systems were very close at 70% and 69%, respectively. The main reason the disc brake/caliper loading system scored poorly compared to the other pump systems is due to its safety score. Safety is the most heavily weighted category in our selection matrix; referring to our thermal analysis, we all agreed that the disc braking system, while potentially effective, was the least safe of the three systems due to the heat generated in the rotors as well as the high rotational velocity of the exposed rotors. Based on the results of our selection matrix, we have decided to eliminate the disc
brake/caliper loading system and focus on the pump loading systems. We will need to further consider each pump system before making a final selection, but additional detail design and analysis can continue in the meantime since both systems are very similar.

Table 6. Pugh Matrix for Selecting the Braking System

<table>
<thead>
<tr>
<th>Category</th>
<th>Weighting</th>
<th>Disc Brake</th>
<th>Dual Pump</th>
<th>Single Pump</th>
</tr>
</thead>
<tbody>
<tr>
<td>Safety</td>
<td>5</td>
<td>2</td>
<td>4</td>
<td>3</td>
</tr>
<tr>
<td>Cost</td>
<td>4</td>
<td>4</td>
<td>2</td>
<td>3</td>
</tr>
<tr>
<td>Simplicity of Operation</td>
<td>2</td>
<td>5</td>
<td>5</td>
<td>5</td>
</tr>
<tr>
<td>Maintenance</td>
<td>3</td>
<td>2</td>
<td>4</td>
<td>4</td>
</tr>
<tr>
<td>Simplicity of Design</td>
<td>4</td>
<td>2.5</td>
<td>3</td>
<td>3</td>
</tr>
<tr>
<td>Simplicity of Manufacturing</td>
<td>3</td>
<td>3</td>
<td>3</td>
<td>3</td>
</tr>
<tr>
<td>Likelyhood of Failure</td>
<td>4</td>
<td>3</td>
<td>4</td>
<td>4</td>
</tr>
<tr>
<td><strong>Total:</strong></td>
<td><strong>73</strong></td>
<td><strong>87</strong></td>
<td><strong>86</strong></td>
<td></td>
</tr>
<tr>
<td><strong>Continue?</strong></td>
<td>No</td>
<td>Yes</td>
<td>Yes</td>
<td></td>
</tr>
</tbody>
</table>
3.4 Loading System Design Refinement

After further design development and research into hydraulic gear pump options, we have refined the loading system design and selected a single hydraulic pump. To simplify the overall system, we removed the transaxle from the engine. The bellhousing engine mount can double as a mounting surface for a hydraulic pump, and a coupling can be used to directly connect the pump to the engine’s flywheel.

![Diagram of Hydraulic Loading System](image)

Figure 12. Diagram of Hydraulic Loading System

Figure 12 shows a layout diagram of the hydraulic loading system. The pump builds pressure in the system, which gets regulated by the needle valve. The amount of pressure on the high pressure side of the circuit, and the flow rate through the pump (which is based on the rpms of the engine) determine how much load the engine sees. The system also includes a filter, to prevent any particulate from getting to the pump, and a relief valve to prevent the pressure from building too high. A heat exchanger is shown to cool the hydraulic system; the function of this could also be accomplished with an oversized fluid reservoir.

3.5 Design Development Summary

The final design layout and components are critically dependent on the type of loading system and its implementation. Since design development has been largely focused on analysis and selection of an appropriate loading system, detail designs of engine and subsystem mounting and integration was delayed. Now that the loading system is narrowed down, final component locations and mounts will be defined.
Final Design

3.6 Design Overview

Figure 13 shows a mockup of the entire test stand, including casters, gauges, indicator lights, and switches, but excludes wiring and plumbing.
3.7 Detaled Design Description

The following sub-sections overview different components of the final design.

3.7.1 Frame Design

Figure 14. Exploded View of Frame Components

The main parts of the Test Stand Frame are shown in Figure 14. More detail is shown in Drawing 2-0, and its following part drawings, in the Appendix.
3.7.2 Engine Mounting

The engine is supported from the bellhousing and the side engine mounts near the front of the engine. The bellhousing mount will also support and locate the hydraulic pump. Both mounts will span the gap between the main rails of the frame and have a pair of polyurethane discs at each mount point for vibration isolation. The engine is offset slightly to one side in relation to the frame to reduce how much the exhaust hangs out past the edge of the frame.

Figure 15. Exploded View of Engine, Mounting Supports, and Vibration Dampers
3.7.3 User Interface

Figure 16. User View of Control Panel Interface

Figure 16 shows a perspective view of the user interface. A layout description of the gauges, switches, and indicator lights can be found on Drawing 2-5b in the Appendix. The Cruise Throttle Lever has the ability to maintain and tune the throttle position for running engine tests.
3.7.4 Loading System

The final loading system includes a gear pump, ball valve, needle valve, over-pressure valve, oil filter, 10 gallon reservoir, pressure gauge, temperature gauge, and hydraulic oil radiator. Some of these components can be seen in Figure 17, above. All of the components are sized to allow loading up to a maximum pressure of 3000 psi and a maximum engine speed of 4000 rpm. Under normal specified requirements, the hydraulic oil radiator should allow for continuous loading of the engine. The ball valve allows for bypassing the needle valve while still pumping through the oil cooler, when loading is not required.
3.8 Analysis Results

Sub-sections below summarize significant analysis results, with supporting calculations available in the appendix.

3.8.1 Frame Static Loading

The frame will be constructed primarily from 2” x 2” x 0.120” mild steel square tubing. Simple static deflection analysis, which can be found in the Appendix, shows a deflection of about 0.050” at the center point of the main frame rails. This analysis was done using a design load of 1000 lbs and, for a worst case scenarios situation, the full length of the frame rails was used excluding any other supports.

3.8.2 Engine Vibration

Without knowing vibrational properties of the engine, it is impossible to accurately analyze the vibrational behavior of the final design. The engine is ridged mounted to cross-members, which are vibration isolated from the frame using polyurethane discs, similar to automotive body mounts.

Selecting a polyurethane mount with a proper durometer is also a challenge, however changing this material after real-world testing would not be difficult. Since calculation of vibrational behavior is unknown, we plan on testing and adjusting the damping material characteristics and thickness if necessary.

3.8.3 Pump Sizing and Integration

Preliminary analysis during the design development showed that a relatively small hydraulic gear pump would be capable of meeting the loading demands to facilitate the smog testing requirement. Since the initial loading system selection, we have chosen to simplify the overall system by removing the transaxle and connecting the hydraulic pump directly to the crankshaft at the flywheel. This simplifies pump mounting and coupling, but makes it necessary to consider the speed at which the engine and pump will be operating. Pumps are available for operating conditions up to 4000 RPM, with momentary maximum rated speeds of 6000 RPM. As long as the engine speed is controlled, a direct coupling will offer the simplest and most reliable interface between the engine and the hydraulic pump.

To supplement the initial sizing calculations, a spreadsheet was developed to aid in solving for the operating conditions of the pump. For a given pump displacement, operating speed, and desired power absorption, the corresponding operating pressure is calculated. The calculation also takes into consideration pump backflow and mechanical efficiency. A version is attached in the appendix, for further reference. These equations and pump properties were also used to create operating curves for the enhanced emissions testing requirements, shown in Figure 18 below.
The engine will be supported on one end by a steel plate with standoffs bolted around the flywheel. This plate will also serve as a mounting surface for the pump. A custom shaft coupling will be fabricated to bolt up to the flywheel and will transmit rotational power to the pump.

### 3.8.4 Pump Coupling to Engine

The gear pump will interface with the flywheel on the motor using a jaw type coupling to transmit power. Jaw couplers have multiple “teeth” on each half of the coupler, which interface with a “spider” which transmits power while allowing for misalignment and damping. The spider can be made of rubber, urethane, or metals such as bronze. Some coupler designs use interlocking teeth which engage a spider under compression, while others have aligned teeth with the spacer transmitting all torque through material shear.

The nominal torque of the engine at 14 H.P. and 1200 RPM is 735 lb-in. Coupling manufacturers typically suggest a service factor of 1.5, making the design torque 1103 lb-in.
Although we looked into off-the-shelf coupling options, we chose to design our own because of the unusual flywheel bolt pattern and the close spacing requirements between the engine and pump. We chose 1045 Steel for the coupling halves, with a 6061 aluminum spider. The spider was designed to be in shear, with the teeth non-interfering. This way, if the coupling were to fail, the pump will become mechanically disconnected from the engine.

Analysis was conducted in Solidworks on the coupling assembly. Since only static analysis was used, the design torque of 1103 lb-in was more than doubled to 2500 lb-in, to give a decent margin of safety for the simplified analysis. Under a 2500 lb-in static torque, the final coupler design had a safety factor of 6. Material failure would occur along the shear plane of the spider between the two couplers, as desired.

![Solidworks FEA Analysis Results for Coupler Design, Factor of Safety](image)

### 3.8.5 Hydraulic Thermal Analysis

The hydraulic system must be able to load the engine with up to 14 horsepower to facilitate the enhanced smog testing requirement. Ultimately all the energy absorbed from the engine will go into heating up the hydraulic fluid. This heat load must be considered, and planned for in the design of the hydraulic system.

Since the test should take no more than about two minutes to perform, the hydraulic fluid will never reach a steady-state operating temperature. The transient nature of this system makes sizing of an appropriate heat exchanger a challenge. Central Coast Bearing, a local hydraulic equipment supplier, suggested that with a large enough reservoir the heat exchanger may not even be necessary. Space for a large reservoir is relatively limited on the test stand, however the heat capacity of the hydraulic fluid in the reservoir and heat loss from the tank should be considered. Below shows the effect of reservoir size on the rate of temperature rise of the hydraulic fluid. Note that the final design utilizes a 10 gallon hydraulic reservoir.
Table 7. Rate of Hydraulic Fluid Temperature Rise for Varying Reservoir Sizes with a 14 HP Load

<table>
<thead>
<tr>
<th>Hydraulic Fluid</th>
<th>Tdot (°C/min)</th>
<th>Tdot (°F/min)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Gallons Kg</td>
<td></td>
<td></td>
</tr>
<tr>
<td>5.0 16.9</td>
<td>19.5</td>
<td>32.5</td>
</tr>
<tr>
<td>7.5 25.3</td>
<td>13.0</td>
<td>21.6</td>
</tr>
<tr>
<td>10.0 33.7</td>
<td>9.7</td>
<td>16.2</td>
</tr>
<tr>
<td>15.0 50.7</td>
<td>6.5</td>
<td>10.8</td>
</tr>
<tr>
<td>20.0 67.4</td>
<td>4.9</td>
<td>8.2</td>
</tr>
</tbody>
</table>

Heat loss from the reservoir would include conduction into the frame, convection from the sides of the tank, and radiation to the surroundings. Conduction to the steel frame was ignored, as thermal resistance between the tank and frame would likely be high. Convection was considered from all four vertical sides of the tank, and radiation to the surroundings was considered for the two sides facing away from the test stand. Calculations, in the appendix, showed a total heat loss of about 200 watts due to convection and radiation, assuming the tank sides are 150 °F and the surroundings are at 70 °F. Since this is approximately two percent of the energy gained from the pump at full load, heat loss from the tank can be ignored. It is worth noting that it will take over 50 times longer for the hydraulic oil to passively cool in the tank than to heat up through engine loading.

In order to allow for repeat loading without full system cool-down, a hydraulic fluid heat exchanger is included in the final design. Hydraulic radiator suppliers simplify sizing by specifying the heat load capacity and acceptable flow rate range for a heat exchanger with a specified air velocity and oil-to-air temperature difference. We chose the Hayden 1268 single-pass oil cooler. The radiator features all copper pipe and aluminum fin construction, along with turbulators inside the tubes for increased cooling effectiveness. This allows a small-sized radiator which meets our maximum flow requirements of 20 GPM, with adequate cooling capacity for near-continuous engine loading. Dimensions and specifications for this heat exchanger are shown in Figure 20, below. Since the hydraulic heat exchanger will be placed after the needle valve, high pressure is not a factor.
### Heavy Duty Oil Cooler Specifications

<table>
<thead>
<tr>
<th>PART #</th>
<th>MOUNTING KITS QTY.</th>
<th>P/N</th>
<th>OVERALL SIZE: HEIGHT, LENGTH, THICKNESS</th>
<th>NUMBER OF ROWS</th>
<th>NUMBER OF PASSES</th>
<th>FITTING SIZE FTP</th>
<th>GPM FLOW RANGE</th>
<th>@2500 AIRFLOW @ 100°F OIL TO AIR</th>
<th>PACKED WEIGHT IN POUNDS</th>
</tr>
</thead>
<tbody>
<tr>
<td>1268</td>
<td>2</td>
<td>288</td>
<td>12-3/4 x 18 x 1-1/2</td>
<td>1</td>
<td>1</td>
<td>3/4</td>
<td>3 – 24</td>
<td>21,200 – 31,200</td>
<td>8</td>
</tr>
</tbody>
</table>

**Figure 20. Hayden 1268 Heavy Duty Oil Cooler Specifications**

### 3.9 Cost Analysis
Table 8 through Table 11, below, summarize the components necessary for each subsystem of the diagnostic test stand. The last two columns, Total 1 and Total 2, list the total cost estimates for each subsystem. Total 1 accounts for parts and materials that we already have or may be able to get at a discount through contacts with local suppliers and represents the estimated minimum cost. Total 2 assumes the full list price for materials and components and represents the estimated maximum cost. Table 12 lists the minimum and maximum projected cost for the entire test stand.
Table 8. Component and price list for subsystems related to the engine

<table>
<thead>
<tr>
<th>Subsystem</th>
<th>Components</th>
<th>$</th>
<th>Quantity</th>
<th>Total 1</th>
<th>Total 2</th>
</tr>
</thead>
<tbody>
<tr>
<td>Engine</td>
<td>3-1/4&quot;x12&quot; 1045 Rod</td>
<td>$62.00</td>
<td>1</td>
<td>$62.00</td>
<td>$62.00</td>
</tr>
<tr>
<td>Radiator</td>
<td></td>
<td>$100.00</td>
<td>1</td>
<td>$100.00</td>
<td>$100.00</td>
</tr>
<tr>
<td>Fuel Cell</td>
<td></td>
<td>$45.00</td>
<td>1</td>
<td>$0.00</td>
<td>$45.00</td>
</tr>
<tr>
<td>Fuel Pump</td>
<td></td>
<td>$60.00</td>
<td>1</td>
<td>$60.00</td>
<td>$60.00</td>
</tr>
<tr>
<td>Serpentine Belt</td>
<td></td>
<td>$25.00</td>
<td>1</td>
<td>$0.00</td>
<td>$25.00</td>
</tr>
<tr>
<td>Cooling Fans</td>
<td></td>
<td>$60.00</td>
<td>2</td>
<td>$120.00</td>
<td>$120.00</td>
</tr>
<tr>
<td>Wire</td>
<td></td>
<td>$30.00</td>
<td>1</td>
<td>$0.00</td>
<td>$30.00</td>
</tr>
<tr>
<td>Misc Electrical Components</td>
<td></td>
<td>$20.00</td>
<td>1</td>
<td>$20.00</td>
<td>$20.00</td>
</tr>
<tr>
<td><strong>Totals</strong></td>
<td></td>
<td></td>
<td></td>
<td><strong>$362.00</strong></td>
<td><strong>$462.00</strong></td>
</tr>
</tbody>
</table>

Table 9. Component and price list for the test stand frame

<table>
<thead>
<tr>
<th>Subsystem</th>
<th>Components</th>
<th>$</th>
<th>Quantity</th>
<th>Total 1</th>
<th>Total 2</th>
</tr>
</thead>
<tbody>
<tr>
<td>Frame</td>
<td>2&quot;x2&quot;x0.120&quot; Square Tubing</td>
<td>$4.00</td>
<td>60</td>
<td>$0.00</td>
<td>$240.00</td>
</tr>
<tr>
<td></td>
<td>3/8&quot; Steel Plate</td>
<td>$18.00</td>
<td>2.75</td>
<td>$0.00</td>
<td>$49.50</td>
</tr>
<tr>
<td></td>
<td>1/4&quot; Steel Plate</td>
<td>$13.00</td>
<td>3</td>
<td>$0.00</td>
<td>$39.00</td>
</tr>
<tr>
<td></td>
<td>1&quot;x12&quot; Steel Rod</td>
<td>$4.00</td>
<td>1</td>
<td>$0.00</td>
<td>$4.00</td>
</tr>
<tr>
<td></td>
<td>1/8&quot;x1-1/2&quot;x1-1/2&quot; Angle Iron</td>
<td>$2.00</td>
<td>6</td>
<td>$0.00</td>
<td>$12.00</td>
</tr>
<tr>
<td>Casters</td>
<td></td>
<td>$15.00</td>
<td>4</td>
<td>$0.00</td>
<td>$60.00</td>
</tr>
<tr>
<td>Misc Hardware</td>
<td></td>
<td>$50.00</td>
<td>1</td>
<td>$50.00</td>
<td>$50.00</td>
</tr>
<tr>
<td><strong>Totals</strong></td>
<td></td>
<td></td>
<td></td>
<td><strong>$50.00</strong></td>
<td><strong>$454.50</strong></td>
</tr>
</tbody>
</table>

Table 10. Component and price list for the hydraulic loading system

<table>
<thead>
<tr>
<th>Subsystem</th>
<th>Components</th>
<th>$</th>
<th>Quantity</th>
<th>Line Total 1</th>
<th>Line Total 2</th>
</tr>
</thead>
<tbody>
<tr>
<td>Hydraulics</td>
<td>Gear Pump</td>
<td>$269.00</td>
<td>1</td>
<td>$269.00</td>
<td>$269.00</td>
</tr>
<tr>
<td></td>
<td>Hydraulic Hose</td>
<td>$2.99</td>
<td>10</td>
<td>$29.90</td>
<td>$29.90</td>
</tr>
<tr>
<td></td>
<td>Hydraulic Fittings</td>
<td>$10.00</td>
<td>8</td>
<td>$80.00</td>
<td>$80.00</td>
</tr>
<tr>
<td></td>
<td>Flow Control Valve</td>
<td>$45.00</td>
<td>1</td>
<td>$45.00</td>
<td>$45.00</td>
</tr>
<tr>
<td></td>
<td>Hydraulic Oil Filter</td>
<td>$40.00</td>
<td>1</td>
<td>$40.00</td>
<td>$40.00</td>
</tr>
<tr>
<td></td>
<td>Hydraulic Reservoir</td>
<td>$60.00</td>
<td>1</td>
<td>$0.00</td>
<td>$60.00</td>
</tr>
<tr>
<td></td>
<td>Hydraulic Oil Cooler</td>
<td>$100.00</td>
<td>1</td>
<td>$100.00</td>
<td>$100.00</td>
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<td></td>
<td></td>
<td><strong>$718.90</strong></td>
<td><strong>$822.17</strong></td>
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Table 11. Component and price list for the test stand user interface

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<th>Quantity</th>
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<th>Total 2</th>
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<td></td>
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<td>$422.07</td>
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Table 12. Total projected minimum and maximum cost for materials and components to complete the test stand

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<th>Project Total</th>
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<tr>
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<td>$1,421.97</td>
<td>$2,160.74</td>
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3.10 Maintenance and Repair

Most of the necessary upkeep for the engine test stand is effectively automotive engine maintenance. Due to the light duty use, necessary maintenance is relatively infrequent. Since the final test stand will be used by the San Luis Obispo High School auto shop, regular maintenance should not be an issue, and can even be used as a learning opportunity for less experienced students.

3.10.1 Engine Maintenance

The 3.8L 3800 Series V6 Engine will require regular standard automotive maintenance as recommended by the manufacturer. Unfortunately, all service recommendations are based on the vehicle’s miles, and therefore will not correlate closely with the test stand’s use. Compared to the use and demands on a normal vehicle, the test stand engine will undergo very light duty use, requiring minimal maintenance.

Engine coolant and oil levels should be checked and topped off every month. The engine oil and filter should be changed every 12 months. The accessory belt should be inspected for cracking and wear every 12 months. The intake air filter should be inspected and cleaned every 6 months.
3.10.2 Hydraulic System Maintenance

Once the hydraulic system is installed, load tested, and leak-checked, it should be almost maintenance-free for the life of the system. Ideally, the hydraulic fluid level should be checked and topped off every month, with fluid and filter replacement every 24 to 36 months.

To avoid leaks, hydraulic line fittings should be checked and tightened every 6 months. Rubber hydraulic lines should be inspected for ballooning and replaced as necessary.
Chapter 4 - Product Realization

Figure 21. Completed Test Stand on Display at the Senior Project Expo

The major components we manufactured include the following:

- Frame
- Engine Mounts
- Fuel and Hydraulic Tanks
- User Interface Panel
- Coupling
- Miscellaneous Components
4.1 Frame

The frame was built using 2" x 2" x 0.120" mild steel square tubing. As seen in Figure 22 above, the joints of each member were joined together using TIG welds.

Holes were drilled in the frame for the engine mount cross-members and 1" x 0.25" steel cylindrical supports were welded inside for additional strength to prevent the tube walls from collapsing.
Figure 24. Frame supports welded into frame rail

4.2 Engine Mounts

Figure 25. Test fitting bellhousing mounting pattern

To make the bellhousing mount, we started by laser cutting an MDF template to make sure our CAD drawing matched the bolt pattern on the engine. This process was repeated multiple times, fine tuning the dimensions to ensure a precise fit. Once we were satisfied with the geometry, the CAD design was sent to Central Coast Creative cutting to be waterjet cut out of 5/16” mild steel plate. The result can be seen in Figure 26.
The rest of the bellhousing engine mount was constructed from 2” x 2” x 0.25” mild steel square tubing. The square tubing was cut into segments and then welded together using multi-pass TIG welds to form the support arch.

The arch was then welded to the steel plate using shielded metal arc welds.
Six 1" diameter steel standoffs were machined and welded to the inside face of the bellhousing mount. Steel webbing was welded between each standoff to provide additional support.

A piece of 2" x 3" x 3/16" angle iron was TIG welded to the bottom of the bellhousing mount. This part of the mount will carry the vertical load of the engine, as opposed to having the welds around the arch carry it, in shear. The entire assembly was then TIG
welded to the cross-member support. Five class 10.9 M12 x 1.75 x 120 bolts and one class 8.8 M12 x 1.75 x 140 bolt were used to secure the bellhousing mount to the engine.

To make the two front facing engine mounts, we suspended the engine from a hoist over the frame and used cardboard to cut templates. The patterns were then traced onto 5/16” mild steel plate and cut out using an oxy-acetylene torch.

Figure 30. Engine Block Mounts on Accessory Side, and a Braided Grounding Strap
4.3 Fuel and Hydraulic Tanks

The fuel and hydraulic tanks were each constructed from 0.065" stainless steel sheet metal. After holes for the filler cap, sight tube, and supply & return lines were drilled, hose fittings were TIG welded from the inside to create a flush exterior. The stainless steel panels were then TIG welded together to finish the tanks.

Figure 31. Completed stainless steel fuel and hydraulic tanks

4.4 User Interface Panel

The user interface panel was fabricated from stainless steel sheet metal. The sheet was cut to size using a large metal shear and then sent to Central Coast Creative Cutting to have the gauge and indicator holes cut out with a water jet.

Figure 32. User interface panel
4.5 Coupling

The complex geometry of the coupling jaws and the spider required them to be machined using a CNC mill. The jaws were machined from 3” diameter cylindrical 1045 steel stock and the spider was machined from a block of 6061 aluminum.

![Figure 33. Coupling jaws](image1)

![Figure 34. Aluminum spider](image2)

4.6 Miscellaneous Components

Some other components we manufactured as part of the test stand include the throttle lever and electrical box.

The throttle lever was made from two pieces of 0.75” x 0.75” x 0.083” mild steel square tubing with mounting through holes and four pieces of thin mild steel flat bar TIG welded together for the support structure.
The electrical box was formed from the same stainless steel sheet metal used for the user interface panel. A 1:1 scale drawing of the flat pattern was used as a template to cut out the electrical box on the shears. Once the flat pattern was cut, we used a brake to bend the sheet into a box.
4.7 Prototype vs Design

As engineers, we try our best to create designs which can be manufactured as close as possible to our specifications. There will always be unforeseen problems that arise when building a prototype though, especially for the first time. The prototype of our test stand is, for the most part, an accurate representation of our intended design. However, there were a few minor changes which had to be made during manufacturing.

When preparing to trace and cut the front engine mounts, we discovered that the front cross-member support interfered with the engine’s oil pan. This issue was due to the length of the machined standoffs for the bellhousing mount changing, as well as possible inaccuracies in our method of creating the CAD model for the engine. To fix this problem we had to offset the cross-member from the mounting holes in the frame by welding a small section of square tubing onto the members as seen in Figure 37.

![Figure 37. Offset engine mount, to address oil pan interference](image)

Another change we made during manufacturing was the placement of the master battery switch. We initially intended the switch to be located on the user interface panel for easy access and the battery to be located underneath. However, the dimensions of the fuel and hydraulic tanks underneath the user interface panel changed, reducing available space and forcing the battery to relocate to the front of the test stand. As a result, we decided to move the master battery switch to the front as well to minimize the amount of heavy gauge wire between the battery and the starter, as seen in Figure 38 below.
4.8 Recommendations for Future Manufacturing

Part of manufacturing a prototype is learning what works and what doesn’t in the process and making note of what to change the next time around. Due to time constraints and several design iterations, we had to begin manufacturing the test stand before all the details of the hydraulic loading system and exhaust system were finalized. Trying to plumb the hydraulics and exhaust simultaneously before the engine, mounts, and cooling system were in place would’ve been very difficult. As a result, the placement of these systems had to wait until most of the test stand was already finished.

Ideally, if we were to build a second test stand, we would have a better idea of the space constraints, allowing us to buy parts and materials for the hydraulics and exhaust sooner and begin installation of these systems in parallel with other manufacturing tasks.
Chapter 5 - Design Verification Plan (Testing)

Many of the design specifications for this project did not require extensive testing procedures, as they were quantitative and verified through measurements and inspection. The frame strength requirement was calculated for a worst-case scenario during design and the actual weight of the entire test stand (without hydraulic oil) was 875 lb, well below the design goal of 1000 lb.

The project went well over the original cost requirements, but the initial budgeting was done prior to the loading system requirement being added, and was not realistically revised with this addition.

Part of our testing plan involved creating at least four step-by-step tutorials on how to perform different diagnostic procedures and then verifying that each procedure could be successfully performed on the test stand. We wanted to create the tutorials once the test stand was operational in conjunction with our sponsor, Jeff Lehmkuhl; however, we decided to hold off for now since we discovered that the engine would not start on the test stand.

Table 13 below lists the design requirements for the test stand and whether they were satisfied or not.
Table 13. Design Requirement Review for Completion

<table>
<thead>
<tr>
<th>Spec. Number</th>
<th>Parameter Description</th>
<th>Requirement or Target (units)</th>
<th>Requirement Met?</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Frame Strength, load of engine and transaxle</td>
<td>1000 lb</td>
<td>Yes</td>
</tr>
<tr>
<td>2</td>
<td>User Interface contains Oil Pressure, Coolant Temp, Ignition-On Light, Tachometer, Check Engine Light, OBD DLC, Ignition Switch</td>
<td>Yes/No</td>
<td>Yes</td>
</tr>
<tr>
<td>3</td>
<td>User Interface Includes a Work Surface Between 36° and 46° off the ground, at least 20&quot; wide and 10&quot; deep.</td>
<td>Yes/No</td>
<td>Yes</td>
</tr>
<tr>
<td>4</td>
<td>Engine will be mounted at a raised height for ease of access for work. Height of Spark Plugs will be at least 30&quot; above the ground.</td>
<td>Yes/No</td>
<td>Yes</td>
</tr>
<tr>
<td>5</td>
<td>System Capable of Being Maneuvered by a Single Average Person</td>
<td>Yes/No</td>
<td>Yes</td>
</tr>
<tr>
<td>6</td>
<td>Engine Includes Key Switch to Prevent Unauthorized Operation</td>
<td>Yes/No</td>
<td>Yes</td>
</tr>
<tr>
<td>7</td>
<td>Moving Components are effectively caged to prevent injury. Check against OSHA Spec.</td>
<td>Yes/No</td>
<td>No</td>
</tr>
<tr>
<td>8</td>
<td>No Electrical or Mechanical Modifications are Made to the Stock Engine or Engine Wiring Harness.</td>
<td>Yes/No</td>
<td>Yes</td>
</tr>
<tr>
<td>9</td>
<td>Additional Raw Material Costs.</td>
<td>$500</td>
<td>No</td>
</tr>
<tr>
<td>10</td>
<td>Additional Components Cost.</td>
<td>$500</td>
<td>No</td>
</tr>
<tr>
<td>11</td>
<td>Final System will Include Step-By-Step Instructional Tutorials.</td>
<td>4 Tutorials</td>
<td>No</td>
</tr>
<tr>
<td>12</td>
<td>System will have all necessary subsystems mounted on-board (Ex. Fuel, Exhaust, Cooling, Battery, Etc.) for Standalone Operation</td>
<td>Yes/No</td>
<td>Yes</td>
</tr>
<tr>
<td>13</td>
<td>Entire System will not exceed 4' wide x 6.5' long x 5' tall</td>
<td>Yes/No</td>
<td>Yes</td>
</tr>
<tr>
<td>14</td>
<td>System includes loading equipment to simulate engine load at 15 and 25 mph.</td>
<td>Yes/No</td>
<td>Yes</td>
</tr>
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</table>
Chapter 6 - Conclusions and Recommendations

We believe we have successfully fulfilled most of the design requirements for this project and, most importantly, created a useful solution for training students in the San Luis Obispo High School Automotive Technology Program to diagnose and troubleshoot older OBD-I based engines.

Our test stand will better meet the needs of the high school’s automotive technology program than similar products on the market. The capability to perform on-board ASM smog testing without additional equipment is a feature not found on other full size test stands and adds significant value to the project.

Months of careful design and analysis validated our concepts for the test stand and chosen loading system. Static analysis of the frame verified that it would be strong enough to support the engine and all of its sub-systems while fluid and thermal analysis of the loading system helped us select an appropriately sized pump and heat exchanger. Static loading simulations also helped us verify that our coupling design was adequate for the power transmission requirements with a large factor of safety.

While we were disappointed that the engine would not start after the fuel, power, cooling, and electronic sub-systems were in place, we want to see the project through and are planning to continue working with Jeff Lehmkuhl beyond senior project to get the engine running and finish the diagnostic tutorials for the test stand. We hypothesize that the PCM has several conditions which must be met before it will allow the engine to start. None of us are well versed in troubleshooting automotive electrical systems, so we believe our best option is to let Jeff and his students diagnose the problem before we continue.
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Appendix A – Project Sponsors

Poly Diagnostic Solutions
Jeff Lewis - Cody Scott - Todd Smith
Sponsor: Jeff Lehmkuhl

Mechanical Engineering
Advisor: Kim Shollenberger

Mike & Kim Cook

NAPA

Marshall Scott

Wendy Söldeblom

Central Coast Creative Cutting
Ian Goodyear

CBI
Roy Coffman

Michelle & Ken Tasseff
Appendix C – Brake Rotor Thermal Equations and Calculations

\[ \Delta T = \frac{H}{C_p W} \]  \hspace{1cm} (16-54)

\( H \) = Rate of braking energy absorbed, Btu/sec
\( \Delta T \) = Rate of temperature rise, °F/sec
\( C_p \) = Specific heat capacity, BTU/(lb \( m \) °F)
\( W \) = Mass of brake disk, lb \( m \)

\[ H_{\text{loss}} = h_{CR} A (T - T_\infty) = (h_r + f_v h_c) A (T - T_\infty) \]  \hspace{1cm} (16-57)

\( H_{\text{loss}} \) = Rate of energy loss, Btu/s
\( h_{CR} \) = Overall coefficient of heat transfer, Btu/(in \( ^2 \) s °F)
\( h_r \) = Radiation component of \( h_{CR} \), Btu/(in \( ^2 \) s °F) Fig.16-24a
\( h_c \) = Convective component of \( h_{CR} \), Btu/(in \( ^2 \) s °F) Fig.16-24a
\( f_v \) = Ventilation factor Fig.16-24b
\( T \) = Disk temperature, °F
\( T_\infty \) = Ambient temperature, °F

---

Figure 16-24a

---

Figure 16-24b
Sample calculations for the temperature rise in a disc brake rotor for the specified parameters.

| Room Air Temp (°F) | 60 |
| Braking Power (HP) | 6 |
| Braking Power (BTU/s) | 4.241 |
| Rotor Mass (lbm) | 10 |
| Rotor cp (Btu/lbm °F) | 0.108 |
| Rotor Diameter (in) | 13 |
| Rotor Surface Area (in²) | 265.5 |
| Ventilation Factor fv | 0 |

Assuming 2-sided disc, no circ, no addl area
From Fig 16-24b

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<tr>
<th>Time (s)</th>
<th>Rotor Temp (°F)</th>
<th>T-T∞∞ (°F)</th>
<th>Braking E Added</th>
<th>H.T. E Dissipated</th>
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<th>Temp Increase</th>
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<td>0.088</td>
<td>4.152</td>
<td>196.1</td>
<td>3.84</td>
</tr>
</tbody>
</table>
\[ \delta_c = \frac{F_a}{24EI} \left( 3L^2 - 4a^2 \right) \]

\[ = \frac{(250 \text{ lb})(15 \text{ in})}{24(29 \times 10^6 \text{ psi})(0.534 \text{ in}^4)} \left[ 3(48 \text{ in}^2) - 4(15 \text{ in})^2 \right] \]

\[ = 0.043 \text{ in} \]

**2" x 2" x 0.120"**

**MILD STEEL SQUARE TUBING**

\[ E_{\text{steel}} = 29 \text{ MSi} \]

\[ I = \frac{b^4 - a^4}{12} \]

\[ = \left( \frac{2 \text{ in}^4}{12} \right) - \left[ \frac{2 \text{ in}^4 - 2(0.4 \text{ in})^4}{12} \right] \]

\[ I = 0.534 \text{ in}^4 \]

\[ L = 48 \text{ in} \]

\[ a = 15 \text{ in} \]

**DESIGN LOAD = 1000 lb**

\[ F = \frac{\text{DESIGN LOAD}}{2 \text{ BEAMS}} \cdot \frac{1 \text{ BEAM}}{2 \text{ MOUNTS}} \]

\[ F = \frac{1000 \text{ lb}}{4} \]

\[ F = 250 \text{ lb} \]
Appendix E – Hydraulic Gear Pump Power Calculations

Calculating Power into Prince SP20 Hydraulic Gear Pump
(northernbkl.com, Item# 106008)

MFR Specs:
Displacement 1.22 cu. in.
Max Flow @ 1500 RPM: 7.49 gpm
Max Flow @ 3500 RPM: 17.76 gpm
Max PSI: 3,000
Max RPM: 3,500

Based on Displacement, Flow @ 1500 rpm, & Flow @ 3500 rpm:

\[
Q_{(gpm)} = 0.00556 \left( \frac{5}{(\text{rpm})} \right) + 0.00071
\]

Below Data from Prince Catalog:

\[
Q_{(gpm)} = 0.00518 \left( \frac{5}{(\text{rpm})} \right) + 0.22714
\]

Fluid Power:

Power = Pressure \cdot Q \cdot \text{flow rate}

Ex. 

Pressure = 2500 psi \cdot \left( \frac{144\, \text{in}^2}{1\, \text{ft}^2} \right) = 360,000 \, \text{lbf/ft}^2

\[ Q = \frac{7.5 \, \text{gpm}}{7.49 \, \text{gal}} \left( \frac{1 \, \text{min}}{60 \, \text{sec}} \right) = 0.016912 \, \text{ft}^3/\text{s} \]

Power = (360,000 \, \text{lbf/ft}^2) \cdot (0.016912 \, \text{ft}^3/\text{s}) \cdot (\frac{1\, \text{hp} \cdot \text{sec}}{550 \, \text{ft} \cdot \text{lbf}})

Power = 11.07 \, \text{HP} \quad \text{(Fluid power)}

General Equation for \, \text{psi}, \, \text{gpm} \, \text{input} \rightarrow \text{Fluid power}

\[
\text{Power (HP)} = \left[ P_{(\text{psi})} \left( \frac{144}{(\text{in})} \cdot Q_{(\text{gpm})} \left( \frac{1}{7.49 \, \text{gal}} \right) \left( \frac{1}{60 \, \text{sec}} \right) \right) \left( \frac{1}{550} \right) \right] \quad \text{[HP]}
\]

\[
W_{\text{mech}} = W_{\text{fluid}} \cdot \frac{1}{\eta_{\text{pump}}}
\]

Assuming \eta_{\text{pump}} = 85% \%

\[
W_{\text{mech}} = 13.02 \, \text{HP}
\]
## Appendix F – Operating Curves for Hydraulic Pump

### Excel Operating Curve Calculations:

<table>
<thead>
<tr>
<th>Pump Specifications</th>
<th>Haldex Gear Pump</th>
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<tbody>
<tr>
<td>Displacement:</td>
<td>1.159 cu. ln.</td>
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<tr>
<td>Max PSI:</td>
<td>3000</td>
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<tr>
<td>Max RPM:</td>
<td>4000</td>
</tr>
<tr>
<td>η:</td>
<td>92%</td>
</tr>
</tbody>
</table>

- **Flowrate Fcn:** $Q(gpm) = 0.0049 \times \text{rpm}$
- **Power Eqn:** $\text{Power} = (\text{P(psi)} \times 144 \text{in}^2/\text{ft}^2 \times Q(gpm) \times \text{rpm}) \times (1 \text{ ft}^3/7.48 \text{ gal}) \times (1 \text{ min/60 sec}) \times (1 \text{ hp/s/550 ft-lbf})$
- **Engine Power:** $\text{Wmech} = \text{Wfluid} \times (1/\eta \text{ pump})$

<table>
<thead>
<tr>
<th>Speed (RPM)</th>
<th>Flow (gpm)</th>
<th>Pressure for given Speed (psi)</th>
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</tr>
<tr>
<td>500</td>
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<td>750</td>
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<td>1000</td>
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<td>508</td>
</tr>
<tr>
<td>5000</td>
<td>24.5</td>
<td>483</td>
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</tbody>
</table>

Page 55
Operating Curves for Enhanced Emissions Test Requirements:

Hydraulic Loading System - Enhanced Emissions Operating Parameters

- MAX
- 7.5 HP
- 12 HP

P = 3.86*10^6*S^-1

P = 2.41*10^6*S^-1

Hydraulic Pump Flow Rate (gal/min)

Engine Speed, S (RPM)
General Pump Operating Curves, Absorbed Power for Different Speeds & Pressures
Appendix G – Temperature Rise of Hydraulic Fluid

Rate of Temperature Rise of Hydraulic Fluid (5 gal.)

\[ V_{\text{fluid}} = \frac{5 \text{ gal.} \cdot 1 \text{ ft}^3}{7.48 \text{ gal.} \cdot 35.31 \text{ ft}^3} = 0.0189 \text{ m}^3 \]

\[ m_{\text{fluid}} = V_{\text{fluid}} \cdot \rho_{\text{fluid}} = 0.0189 \text{ m}^3 \cdot 890 \text{ kg/m}^3 = 16.85 \text{ kg} \]

\[ P_{\text{in}} = 14 \text{ HP} \]

\[ q_{\text{in}} = 14 \text{ HP} \cdot \frac{0.7457 \text{ KW}}{1 \text{ HP}} = 10.44 \text{ KW} \]

\[ \dot{t}_{\text{fluid}} = \frac{q_{\text{in}}}{m \cdot \rho} = 10.44 \text{ Ks} \cdot \left( \frac{1}{16.85 \text{ kg} \cdot 1.909 \text{ KS/kg-K}} \right) \cdot \frac{60 \text{ s}}{\text{min}} \]

\[ \dot{t}_{\text{fluid}} = 19.5 \text{ °C/min} = 32.5 \text{ °F/min} \]

The Hydraulic Fluid would reach 150°F in about 25 minutes of operation

\[ q_{\text{in}} = 14 \text{ HP} \cdot \frac{2544 \text{ Btu/HR}}{1 \text{ HP}} = 35,620 \frac{\text{ Btu}}{\text{HR}} \]
Appendix H – Heat Loss from Hydraulic Tank

Convection Heat Loss from Reservoir

Assumptions: Tank sides at 150°F, \( T_{amb} = 70°F \)

Tank Dimensions: 16" W x 6" L x 12" H

\[
Ra_L = \frac{g \beta \Delta T L^3}{\nu \alpha} = \frac{(9.81 \text{ m/s}^2)(\frac{1}{316.5 \text{ k}})(339 \text{ k} - 294 \text{ k})(0.305 \text{ m})^3}{(15.89 \times 10^6 \text{ m}^2/\text{s})(22.5 \times 10^{-6} \text{ m}^2/\text{s})}
\]

\[Ra_L = 110.6 \times 10^6\]

For a Vertical Plate

\[
\overline{N_u} = \frac{\overline{h} L}{k} = \left[ 0.825 + \frac{0.387 \overline{Ra}^{1/6}}{[1 + (0.492/Pr)^{9/16}]^{8/27}} \right]^{2/3}
\]

\[
\overline{N_u} = \frac{\overline{h} (0.305 \text{ m})}{2.63 \times 10^{-3} \text{ W/mK}} = 62.9
\]

\[
\overline{h} = 5.42 \text{ W/m}^2 \text{ K}
\]

\[
q = \overline{h} A (T_H - T_C) = 5.17 \text{ W/m}^2 \text{ K} (0.344 \text{ m}^2) (339 \text{ K} - 294 \text{ K})
\]

\[q = 80.0 \text{ Watts}\]

Radiation Heat Loss, 2 sides

Assume: \( E = 0.5 \) (oxidized Al)

\[A_s = 0.170 \text{ m}^2\]

\[
q_{rad} = E \sigma A_s (T_s^4 - T_{sw}^4)
\]

\[
= (0.5)(5.67 \times 10^{-8} \text{ W/m}^2 \text{ K}^4)(0.170 \text{ m}^2)(339^4 - 294^4)
\]

\[q_{rad} = 27.6 \text{ Watts}\]

\[q_{tot} = 107.6 \text{ Watts}\]
Excel Calculation of Tank Heat Loss, with Updated Tank Dimensions

Heat Loss from Tank Sides

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<th>Inches</th>
<th>meters</th>
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<td>Length</td>
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<tr>
<td>Height</td>
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Convection

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<td>Nu av.</td>
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<tr>
<td>h average</td>
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Radiation

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<td>q rad. length</td>
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<td>q rad. (2 sides)</td>
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<td>q conv. length</td>
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<td>q conv. (4 sides)</td>
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Total Tank Heat Loss

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Air Prop @ Tfilm

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<td>Mu</td>
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<td>alpha</td>
<td>2E-05 m²/s</td>
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<td>Pr</td>
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Emissivity 0.5
Appendix I – Heat Loss from Hydraulic Radiator

Hydraulic Oil Cooler Load Sizing

Design Load: 12 HP

Conversion: 1 HP = 2,544 BTU/hr

\[
12 \text{ HP} \cdot \frac{2,544 \text{ BTU/hr}}{1 \text{ HP}} = 30,533 \text{ BTU/hr}
\]

Hayden 1268 Oil Cooler Rating:

21,200 - 31,200 Cooling BTU/hr

(at 2500 CFM Airflow, 100°F ITD Oil-to-Air)
Appendix J – Hydraulic Component Specifications

The following sheets are manufacturer specifications for critical hydraulic components.

- **Hydraulic Pump:** Haldex WP09A1B190R03BA103N
- **Hydraulic Radiator:** Hayden 1268 Oil Cooler
- **Needle Valve:** Parker N1200S/1A867
- **Over-Pressure Vlv:** Prince 4HL34
- **Hydraulic Oil Filter:** Napa Gold 1453
- **Electric Fans:** SPAL 0365
W900 DISTRIBUTOR STOCK PUMPS

W900 SERIES PUMPS WITH SAE “A” 2-BOLT MOUNT, 3/4” DIA. SAE “A” STRAIGHT SHAFT, AND SAE STRAIGHT THREAD SIDE PORTS

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W900 SERIES PUMPS WITH SAE “A” 2-BOLT MOUNT, 9 TOOTH SAE “A” SPLINE SHAFT, AND SAE STRAIGHT THREAD SIDE PORTS

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W900 SERIES PUMPS WITH SAE “A” 2-BOLT MOUNT, 11 TOOTH SAE “A” SPLINE SHAFT, AND SAE STRAIGHT THREAD SIDE PORTS

<table>
<thead>
<tr>
<th>DISPLACEMENT IN.</th>
<th>CC</th>
<th>ROTATION</th>
<th>SAE SIDE PORTS</th>
<th>MODEL</th>
<th>CATALOG</th>
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</table>
Designed for rugged heavy duty use, our coolers can be used for many different cooling applications, including transmission, engine, gear box, hydraulic oils and diesel fuel.

Our heavy duty coolers feature brazed copper tubes with aluminum fin construction. Turbulators in every tube provide 3 1/2 times more cooling surface than bare tubing.

Heavy Duty Oil Cooler Specifications

<table>
<thead>
<tr>
<th>PART #</th>
<th>MOUNTING KITS</th>
<th>QTY.</th>
<th>P/N</th>
<th>OVERALL SIZE</th>
<th>NUMBER OF ROWS</th>
<th>NUMBER OF PASSES</th>
<th>FITTING SIZE</th>
<th>GPM FLOW RANGE</th>
<th>COOLING BTU / HR. @2500 AIRFLOW @ 100°F ITD OIL TO AIR</th>
<th>PACKED WEIGHT IN POUNDS</th>
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</thead>
<tbody>
<tr>
<td></td>
<td>A</td>
<td>B</td>
<td>C</td>
<td>D</td>
<td>E</td>
<td>F</td>
<td>G</td>
<td>H</td>
<td>I</td>
<td>J</td>
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<td>TWO-PASS COOLERS</td>
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<td>7,300 – 12,000</td>
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<td>22-1/8 x 24 x 1-1/2</td>
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<td>2</td>
<td>3/4</td>
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<td>36,500 – 61,000</td>
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<td>6,000 – 8,500</td>
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<td>1</td>
<td>1/2</td>
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<td>16,900 – 25,900</td>
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<td>21,200 – 31,200</td>
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<td>3/4</td>
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<td>6 – 48</td>
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<td>7.5 – 60</td>
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<td>1-1/4</td>
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<td>1-1/4</td>
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<td>1</td>
<td>12 – 96</td>
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<td>1-1/2</td>
<td>18 – 144</td>
<td>235,900 – 415,700</td>
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</table>
General Description

Series N needle valves are ideal as speed controls on hydraulic and pneumatic systems where a reverse flow check is not needed. They provide excellent control and a reliable shut-off in a very small envelope.

Operation

A two-step needle allows fine adjustment at low flow by using the first three turns of the adjusting knob. The next three turns open the valve to full flow, and also provide standard throttling adjustments.

Features

- The exclusive “Colorflow” color-band reference scale on the valve stem is a great convenience and time-saver in setting the valve originally and in returning it to any previous setting.
- A simple set screw locks the valve on any desired setting.
- A tamperproof option (T) feature is also available to prevent accidental or intentional adjustment of flow setting.

Specifications

<table>
<thead>
<tr>
<th>Maximum Operating Pressure</th>
<th>Brass: 140 Bar (2000 PSI); except for N1600 brass which is 35 Bar (500 PSI)</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Steel &amp; Stainless: 345 Bar (5000 PSI) for Steel; 207 Bar (3000 PSI) for Stainless Steel and all other sizes</td>
</tr>
<tr>
<td>Material</td>
<td>Body: see ordering code; Knob: Steel - Zinc plated; Needle: 416 Stainless Steel; Stainless Steel Bodies: 303 Stainless Steel</td>
</tr>
<tr>
<td>Operating Temperature</td>
<td>-40°C to +121°C (-40°F to +250°F) Nitrile (standard) and -26°C to +205°C (-15°F to +400°F) Fluorocarbon</td>
</tr>
</tbody>
</table>

Performance Curves

Controlled Flow vs. Pressure Drop

- Models 400 and 620 (#4) Model 200
- Models 400 and 620
- Models 600 and 820
- Model 2020
- Models 1600 and 2000
- Models 1600 and 2000
- Models 1600 and 2000
- Models 1600 and 2000
- Models 1600 and 2000
Flow Control Valves
Series N

Options

N
Series N

Size

Material

Needle
Options

Other
Options

Seal
Compound

Design
Series

NOTE:
Not required when ordering.

Options

Code  Description

Code  Description

Omit NPTF/SAE
★ 8  BSPT
★★ 9  BSPP

★  Code 8 can be used with sizes 400, 600, 800 Steel only

★★ Code 9 can be used with sizes 200, 400, 600, 800, 1200, 1600, 2000

Code  Description

Code  Description

Seal

Code  Description

Omit Nitrile (Standard)
V  Fluorocarbon
  200 - 1020
  1200 - 2000

E  EPR

Series N Brass Valves can be used for both air and oil service.

* Available in 400, 600 and 800 sizes.

<table>
<thead>
<tr>
<th>Model Number</th>
<th>Max Flow LPM (GPM)</th>
<th>Effective Orifice Area Control Flow in.²</th>
<th>Effective Control Flow Cₐ</th>
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<td>11 (3)</td>
<td>0.0102</td>
<td>0.230</td>
</tr>
<tr>
<td>N420</td>
<td>11 (3)</td>
<td>0.0102</td>
<td>0.230</td>
</tr>
<tr>
<td>N400</td>
<td>19 (5)</td>
<td>0.0194</td>
<td>0.443</td>
</tr>
<tr>
<td>N620</td>
<td>19 (5)</td>
<td>0.0194</td>
<td>0.443</td>
</tr>
<tr>
<td>N800</td>
<td>30 (8)</td>
<td>0.0344</td>
<td>0.787</td>
</tr>
<tr>
<td>N820</td>
<td>30 (8)</td>
<td>0.0344</td>
<td>0.787</td>
</tr>
<tr>
<td>N1020</td>
<td>57 (15)</td>
<td>0.0427</td>
<td>0.976</td>
</tr>
<tr>
<td>N1200</td>
<td>57 (15)</td>
<td>0.0427</td>
<td>0.976</td>
</tr>
<tr>
<td>N1220</td>
<td>95 (25)</td>
<td>0.1080</td>
<td>2.470</td>
</tr>
<tr>
<td>N1600</td>
<td>151 (40)</td>
<td>0.2300</td>
<td>5.250</td>
</tr>
<tr>
<td>N1620</td>
<td>151 (40)</td>
<td>0.3070</td>
<td>7.000</td>
</tr>
<tr>
<td>N2000</td>
<td>264 (70)</td>
<td>0.2300</td>
<td>5.250</td>
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<td>N2020</td>
<td>264 (70)</td>
<td>0.3710</td>
<td>8.470</td>
</tr>
<tr>
<td>N2400</td>
<td>379 (100)</td>
<td>0.2300</td>
<td>5.250</td>
</tr>
</tbody>
</table>

Series N Brass Valves

can be used for both air and oil service.

* Available in 400, 600 and 800 sizes.

★  Code 8 can be used

with sizes 400, 600, 800 Steel only

★★ Code 9 can be used

with sizes 200, 400, 600, 800, 1200, 1600, 2000

* Sizes available in Brass.

† Sizes available in Stainless Steel.

Code  Size

200 * 1/8"
400 † 1/4"
420 #4 SAE
600 † 3/8"
620 #6 SAE
800 † 1/2"
820 #8 SAE

* Sizes available in Brass.

† Sizes available in Stainless Steel.
### Flow Control Valves

**Series N**

Inch equivalents for millimeter dimensions are shown in (**).

#### Knob Options

- **Tamperproof Option (Code “T”)** permanently locks knob at desired flow setting by installing a pin in predrilled hole.
- **Finger screw Option (Code “F”)** provides this thumb-screw in place of set screw.

#### Dimensions

<table>
<thead>
<tr>
<th>Model Number</th>
<th>Weight kg (lbs.)</th>
<th>A</th>
<th>B</th>
<th>C</th>
<th>D</th>
<th>E</th>
<th>F</th>
<th>G</th>
<th>H</th>
</tr>
</thead>
<tbody>
<tr>
<td>N200</td>
<td>0.1 (0.3)</td>
<td>39.1 (1.54)</td>
<td>35.3 (1.39)</td>
<td>38.1 (1.50)</td>
<td>19.1 (0.75)</td>
<td>15.7 (0.62)</td>
<td>7.9 (0.31)</td>
<td>19.1 (0.75)</td>
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</tr>
<tr>
<td>N400</td>
<td>0.2 (0.5)</td>
<td>45.5 (1.79)</td>
<td>40.4 (1.59)</td>
<td>50.8 (2.00)</td>
<td>25.4 (1.00)</td>
<td>20.6 (0.81)</td>
<td>10.4 (0.41)</td>
<td>20.6 (0.81)</td>
<td></td>
</tr>
<tr>
<td>N420</td>
<td>0.1 (0.3)</td>
<td>41.4 (1.63)</td>
<td>37.6 (1.48)</td>
<td>50.8 (2.00)</td>
<td>25.4 (1.00)</td>
<td>20.6 (0.81)</td>
<td>10.4 (0.41)</td>
<td>19.1 (0.75)</td>
<td></td>
</tr>
<tr>
<td>N600</td>
<td>0.4 (0.9)</td>
<td>55.4 (2.18)</td>
<td>49.5 (1.95)</td>
<td>63.5 (2.50)</td>
<td>31.8 (1.25)</td>
<td>25.4 (1.00)</td>
<td>12.7 (0.50)</td>
<td>25.4 (1.00)</td>
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</tr>
<tr>
<td>N620</td>
<td>0.2 (0.5)</td>
<td>47.8 (1.88)</td>
<td>42.7 (1.68)</td>
<td>60.5 (2.38)</td>
<td>30.2 (1.19)</td>
<td>25.4 (1.00)</td>
<td>12.7 (0.50)</td>
<td>20.6 (0.81)</td>
<td></td>
</tr>
<tr>
<td>N800</td>
<td>0.6 (1.4)</td>
<td>68.6 (2.70)</td>
<td>61.5 (2.42)</td>
<td>66.5 (2.62)</td>
<td>33.3 (1.31)</td>
<td>31.8 (1.25)</td>
<td>15.7 (0.62)</td>
<td>30.2 (1.19)</td>
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</tr>
<tr>
<td>N820</td>
<td>0.4 (0.9)</td>
<td>56.9 (2.14)</td>
<td>51.1 (2.01)</td>
<td>76.2 (3.00)</td>
<td>38.1 (1.50)</td>
<td>28.4 (1.12)</td>
<td>14.2 (0.56)</td>
<td>25.4 (1.00)</td>
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<tr>
<td>N1020</td>
<td>0.6 (1.3)</td>
<td>68.6 (2.70)</td>
<td>61.5 (2.42)</td>
<td>88.9 (3.50)</td>
<td>44.5 (1.75)</td>
<td>31.8 (1.25)</td>
<td>15.7 (0.62)</td>
<td>30.2 (1.19)</td>
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<tr>
<td>N1200</td>
<td>1.0 (2.3)</td>
<td>85.9 (3.38)</td>
<td>71.4 (2.81)</td>
<td>82.6 (3.29)</td>
<td>41.1 (1.62)</td>
<td>38.1 (1.50)</td>
<td>19.1 (0.75)</td>
<td>35.1 (1.38)</td>
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<tr>
<td>N1220</td>
<td>1.0 (2.3)</td>
<td>85.9 (3.38)</td>
<td>71.4 (2.81)</td>
<td>101.6 (4.00)</td>
<td>50.8 (2.00)</td>
<td>38.1 (1.50)</td>
<td>19.1 (0.75)</td>
<td>35.1 (1.38)</td>
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</tr>
<tr>
<td>N1600</td>
<td>2.1 (4.7)</td>
<td>123.7 (4.87)</td>
<td>106.9 (4.21)</td>
<td>108.0 (4.25)</td>
<td>53.8 (2.12)</td>
<td>44.5 (1.75)</td>
<td>22.4 (0.88)</td>
<td>47.8 * (1.88)</td>
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</tr>
<tr>
<td>N1620</td>
<td>2.1 (4.7)</td>
<td>130.8 (5.15)</td>
<td>114.0 (4.49)</td>
<td>108.0 (4.25)</td>
<td>53.8 (2.12)</td>
<td>57.2 (2.25)</td>
<td>28.4 (1.12)</td>
<td>47.8 * (1.88)</td>
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</tr>
<tr>
<td>N2000</td>
<td>2.9 (6.4)</td>
<td>130.0 (5.12)</td>
<td>113.3 (4.46)</td>
<td>108.0 (4.25)</td>
<td>53.8 (2.12)</td>
<td>57.2 (2.25)</td>
<td>28.4 (1.12)</td>
<td>47.8 * (1.88)</td>
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<tr>
<td>N2020</td>
<td>2.9 (6.4)</td>
<td>140.2 (5.52)</td>
<td>123.4 (4.86)</td>
<td>114.3 (4.50)</td>
<td>57.2 (2.25)</td>
<td>69.9 (2.75)</td>
<td>60.5 (2.38)</td>
<td>47.8 * (1.88)</td>
<td></td>
</tr>
</tbody>
</table>

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* = Hex
Hydraulic Valves

Flow Control, Needle, and Check Valves
Brass valves are for pneumatic and medium-pressure hydraulic applications. Steel valves are for high-pressure hydraulic applications.

**FLOW CONTROL VALVES**
Provide precision flow control in 1 direction, and automatically allow unrestricted free flow in opposite direction. 2-step needle allows fine adjustment at low flow.

**NEEDLE VALVES**
Used as speed controls where a reverse flow check valve is not needed. May be used to control flow in both directions.

**CHECK VALVES**
Provide free flow in 1 direction and dependable shutoff in the opposite direction. Cracking pressure: 5 psi.

<table>
<thead>
<tr>
<th>Port Size (FPMPT)</th>
<th>Max. Range (gpm)</th>
<th>FLOW CONTROL VALVES</th>
<th>NEEDED VALVES</th>
<th>CHECK VALVES</th>
</tr>
</thead>
<tbody>
<tr>
<td>Brass Valves 5000 psi</td>
<td>¼-27</td>
<td>3.00</td>
<td>1.54</td>
<td>F2006</td>
</tr>
<tr>
<td>½-18</td>
<td>5.00</td>
<td>1.79</td>
<td>F4006</td>
<td>1A854</td>
</tr>
<tr>
<td>¾-14</td>
<td>7.50</td>
<td>2.18</td>
<td>F6006</td>
<td>1A855</td>
</tr>
<tr>
<td>Steel Valves 3500 psi</td>
<td>¼-18</td>
<td>5.00</td>
<td>1.79</td>
<td>F4005</td>
</tr>
<tr>
<td>½-18</td>
<td>7.50</td>
<td>2.18</td>
<td>F6005</td>
<td>1A042</td>
</tr>
<tr>
<td>Steel Valves 5000 psi</td>
<td>¼-14</td>
<td>4.00</td>
<td>2.70</td>
<td>F8006</td>
</tr>
</tbody>
</table>

* Height of flow control and needle valves is measured from center of port with adjustment knob fully open.

Metering Valves and Panel Mount Kits
Metering valves allow high precision metering and shutoff of flow. Standard needle allows fine adjustment at low flow using 3 turns of adjusting knob. Exclusive Colorflow® color-band reference scale on valve stem simplifies initial setting of valve and also the return to a previous setting. Panel Mount Kits include washer and nut for easy mounting of metering valves.

<table>
<thead>
<tr>
<th></th>
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<th></th>
<th></th>
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<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>¼ NPT</td>
<td>4 GPM</td>
<td>4DKE6</td>
<td>379.25</td>
<td>0.80</td>
<td>0.60</td>
<td>4DKF3</td>
<td>379.25</td>
<td>4DKF7</td>
<td>379.25</td>
<td>4DKK2</td>
<td>379.25</td>
<td>4DKK6</td>
</tr>
<tr>
<td>½ NPT</td>
<td>8 GPM</td>
<td>4DKF1</td>
<td>498.25</td>
<td>1.00</td>
<td>0.80</td>
<td>4DKF2</td>
<td>498.25</td>
<td>4DKF6</td>
<td>498.25</td>
<td>4DKK1</td>
<td>498.25</td>
<td>4DKK5</td>
</tr>
<tr>
<td>¾ NPT</td>
<td>15 GPM</td>
<td>4DKF9</td>
<td>510.50</td>
<td>1.50</td>
<td>1.00</td>
<td>4DKF4</td>
<td>510.50</td>
<td>4DKK7</td>
<td>510.50</td>
<td>4DKK7</td>
<td>510.50</td>
<td></td>
</tr>
<tr>
<td>Steel Valves 5000 psi</td>
<td>¼-14</td>
<td>3.00</td>
<td>3.38</td>
<td>F12005</td>
<td>1A865</td>
<td>125.95</td>
<td>—</td>
<td>—</td>
<td>—</td>
<td>—</td>
<td>—</td>
<td>—</td>
</tr>
</tbody>
</table>

* Height is measured from center of port with adjustment knob fully open.

Pressure Control Valves
These brass inline valves allow the system to open to tank when system pressure reaches the externally adjustable pressure setting of the control valve.

**PRESSURE CONTROL VALVES**

<table>
<thead>
<tr>
<th>Port Size (NPT)</th>
<th>Max. Flow (gpm)</th>
<th>10 PSI PRESET 15 psi Adjustment Range</th>
<th>Item No.</th>
<th>$ Each</th>
<th>35 PSI PRESET 50 psi Adjustment Range</th>
<th>Item No.</th>
<th>$ Each</th>
<th>90 PSI PRESET 150 psi Adjustment Range</th>
<th>Item No.</th>
<th>$ Each</th>
<th>200 PSI PRESET 250 psi Adjustment Range</th>
<th>Item No.</th>
<th>$ Each</th>
<th>360 PSI PRESET 450 psi Adjustment Range</th>
<th>Item No.</th>
<th>$ Each</th>
</tr>
</thead>
<tbody>
<tr>
<td>¼ NPT</td>
<td>4 GPM</td>
<td>4DKJ3</td>
<td>382.75</td>
<td>4DKJ4</td>
<td>382.75</td>
<td>4DKJ5</td>
<td>382.75</td>
<td>4DKJ6</td>
<td>382.75</td>
<td>4DKJ7</td>
<td>382.75</td>
<td>4DKJ8</td>
<td>382.75</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>½ NPT</td>
<td>8 GPM</td>
<td>4DKJ8</td>
<td>500.50</td>
<td>4DKJ9</td>
<td>510.50</td>
<td>4DKJ10</td>
<td>510.50</td>
<td>4DKJ11</td>
<td>510.50</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

Relief Valves
Control maximum pressure within a hydraulic circuit. No. 6X4A5 is designed for full flow with low pressure drop and constructed of high-strength steel bar stock body with replaceable heat-treated seat.

**RELIEF VALVES**

<table>
<thead>
<tr>
<th></th>
<th></th>
<th></th>
<th></th>
<th></th>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>Single</td>
<td>¼ NPT</td>
<td>15 GPM</td>
<td>1000 to 2500</td>
<td>Prince</td>
<td>R1820H</td>
<td>64.65</td>
</tr>
<tr>
<td>½ NPT</td>
<td>15 GPM</td>
<td>2500 to 5000</td>
<td>Vickers</td>
<td>CS-03-C50</td>
<td>48A48</td>
<td>433.30</td>
</tr>
</tbody>
</table>

No. 6X4A5 is a double differential poppet relief that is used in systems that require cross-over relief protection such as a hydraulic motor.

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Find even MORE at Grainger.com®

3787
Transmission Oil Filter (Gold)

Part Number: FIL 1453
Product Line: NAPA Gold Filters

Attributes:
- Gasket I.D. (Inches) : 2.415"
- Gasket I.D. (mm) : 61.34 mm
- Oil Filter Gasket O.D. (Inches) : 2.729"
- Oil Filter Gasket O.D. (mm) : 69.32 mm
- Oil Filter Gasket Thickness (Inches) : .255"
- Oil Filter Gasket Thickness (mm) : 6.48 mm
- Oil Filter Height (Inches) : 5.157"
- Oil Filter Height (mm) : 130.99 mm
- Oil Filter O.D. (Inches) : 3.67"
- Oil Filter O.D. (mm) : 93.22 mm
- Oil Filter Style : Spin-On Hydraulic Filter
- Oil Filter Thread Size (inches) : 1-12

Features and Benefits:
NAPA Gold Oil Filters Hold 45% More Dirt + Last 30% Longer Than Other Leading Competitive Brands. 32 Micron

Warranty:
NAPA Filters are Covered by a Comprehensive Limited Product Warranty. NAPA Filters May Pay the Reasonable Cost for Parts & Labor to Repair any System Damaged by NAPA Filters Due to a Defect in Design or Material.

Warranty PDF

Important Information:
8x4 Trans

Have questions about this item?
Call your local NAPA AUTO PARTS Store at 800-LET-NAPA (800-538-6272)

Buyer's Guide:
For a list of specific cars that this part is compatible with, please visit our Buyer's Guide.
AMP connector Code 180908
AMP terminal Code 42098-2
Suggested mounting torque: 3(+1/-0) Nm with screw M5
Weight 1,1 Kg. approx.

Connettore AMP Cod. 180908
Terminali AMP Cod. 42098-2
Coppia di serraggio consigliata: 3(+1/-0) Nm con vite M5
Peso 1,1 Kg. circa

PRODUCT FEATURES
CARATTERISTICHE PRODOTTO

Waterproof motor, IP 68
Motore chiuso, IP 68

Long life / Lunga durata

LL and VLL version
Versione LL e VLL

Waterproof connector
Connettori a tenuta stagna

Accessories: all the fixing kits
Acessor: tutti i kit di fissaggio

<table>
<thead>
<tr>
<th>Static pressure</th>
<th>Airflow Portata</th>
<th>Current input</th>
<th>Airflow Portata</th>
<th>Current input</th>
<th>Airflow Portata</th>
<th>Current input</th>
<th>Static pressure</th>
</tr>
</thead>
<tbody>
<tr>
<td>Pressione statica</td>
<td>m³/h</td>
<td>A</td>
<td>CFM</td>
<td>m³/h</td>
<td>A</td>
<td>CFM</td>
<td>mm H₂O</td>
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<tr>
<td>0</td>
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<td>755</td>
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<td>6,5</td>
<td>684</td>
<td>1140</td>
<td>6,5</td>
<td>673</td>
<td>0,1</td>
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<tr>
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<td>6,8</td>
<td>578</td>
<td>1000</td>
<td>6,6</td>
<td>590</td>
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<td>160</td>
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<td>6,7</td>
<td>0</td>
<td>0,7</td>
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<td>6,7</td>
<td>0</td>
<td>0</td>
<td>0,8</td>
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</tr>
</tbody>
</table>

* for OEM applications only. ✓ = Standard feature
* per applicazioni OEM = Caratteristiche standard
✓ = Available upon request
✓ = Disponibile su richiesta

Test voltage 13 V. d.c. - Tensione di prova 13 V. c.c.
1991 Buick Park Avenue

Fig. 3: Fuse Block, Ign Switch, ELC, Heated W/Shield (Grid 8-11)
1991 Buick Park Avenue
Fig. 5: A/C-Heater System (Grid 16-19)
1991 Buick Park Avenue

Fig. 6: DERM, Wipers, Gear Select Switch (Grid 20-23)
1991 Buick Park Avenue
Fig. 9: Cruise, Power Doors (Grid 32-35)
NOTE: All NPT tapped holes are threaded holes.

Upon assembling the tank, weld the 1/8 NPT connections.

Dimensions are provided in inches.