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Project 7

Hydraulically Powered On-Demand Supercharger Boost System

Design Review V
Final Written Report

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I. EXECUTIVE SUMMARY

The USEPA is involved in research of high-efficiency, low emissions engine technologies. The goal of this project is to design, verify, and build an on-demand supercharger, powered by a small hydraulic motor, to produce additional boost for mid-range speeds according to the desired pressure profile for 6 to 7 seconds. The reason for the use of a hydraulic motor is that in order to minimize power deduction from the engine, efforts are being made to utilize pressurized transmission fluid. Current fluid-powered superchargers, known as hydrachargers, do not meet the customer requirements. By accomplishing our goals, we have created an on-demand supercharger that currently does not exist, and could readily be adapted to use with a hydraulic hybrid chassis. The USEPA communicated the driving customer requirements, that it meets the pressure/flow profile and is durable, efficient, and quick to power up. The team developed engineering specifications based on these customer requirements, and then generated multiple prototype concepts and weighed how they each met the specifications.

An Alpha prototype was developed in Design Review #2, and was then modified into the prototype concept, with a Roots type supercharger instead of a centrifugal compressor. This decision was based on the ability for a Roots type supercharger to produce boost at low operating speeds, whereas the centrifugal compressor is most effective at high RPMs. Our prototype and final design were developed to meet the engineering specifications. The manufacturing plan for the prototype was established to streamline the manufacturing process and help the team determine which components can realistically be manufactured before the Design Expo deadline. The out-of-pocket cost for the prototype was $180.13, which includes aluminum plates and pipes, the belt tensioner, and the belt.

Multiple methods were used to help validate the prototype. The first of these methods was simple physical evaluation of the behavior of the system. The next method of testing, which we did not complete, is to run the system using the proper power and speeds. The final method of testing was numerical analysis of the performance of the system. To do this numerical simulation, we created a code in Matlab to put together all of the necessary variables and to use a time-stepping method of evaluating the performance of the system.

Our designs for both our prototype and our final design are well thought out and fully engineered to the best of our team's abilities. We completed all of the necessary research to understand the operation of our system and the requirements on the geometry of our prototype, and chose all parameters and dimensions carefully. The design process led us to a prototype that provides not only for the opportunity to test a full-scale model of the final design, but also for precise manipulation of the displacement of the hydraulic motor in order to fully characterize system performance. However, the approximate calculated efficiency of the prototype on-demand supercharger system is low, 33.6%. This is due to the combined lower efficiencies for both the motor and the supercharger, due to the over-designed nature of our prototype. Our group recognizes that the system can be improved. The prototype we have developed will utilize a supercharger and hydraulic motor that are already in existence to produce the desired pressure profile. For future modifications of the prototype design, improvement can be made with respect to choosing a hydraulic motor and supercharger that can be running at or near its ideal operating point while achieving the desired pressure/flow profile.

II. INTRODUCTION, BACKGROUND & CUSTOMER REQUIREMENTS

The USEPA is involved in research of high-efficiency, low emissions engine technologies. Specifically, there is research being conducted involving clean-burning diesel engines fitted with forced induction technology. The type of forced induction which our project involves is called supercharging, which is normally connected to the engine drive belt system. This system takes in ambient air and pressurizes it into the piston cylinders to improve engine performance.

Current laboratory testing has enabled the use of forced induction at both the low end and high end of motor RPM. These two separate turbochargers have the ability to produce what is known as “boost pressure” that is used to describe the elevation in pressure above the ambient. The ideal case would be to produce a near constant boost pressure throughout the range of engine operation. However, the current attached turbochargers only operate at certain RPM ranges to produce the desired pressure, and leave a gap in the midrange boost output for the engine. Thus, these engines require additional intake pressure (boost) as they transition between low and high speed turbochargers.

The goal of this project is to design, verify, and build an on-demand supercharger, powered by a small hydraulic motor, to produce additional boost for mid-range speeds according to the desired pressure profile for 6 to 7 seconds. Our design should be durable, efficient, and quick to power up. The reason for the use of a hydraulic motor is that in order to minimize power deduction from the engine, efforts are being made to utilize pressurized transmission fluid. Current fluid-powered superchargers, known as hydrachargers, do not meet the customer requirements for both efficiency and performance control. The significance of this project is that if we are able to accomplish our goals, we will have created an efficient on-demand supercharger that could readily be adapted to use with a hydraulic hybrid chassis. Since this type of supercharger does not currently exist, the success of the project could lead to patent application and production use.

The USEPA communicated several customer requirements to the team during initial project discussions. They also outlined a number of engineering specifications which we will need to take into account during the design process. All of the customer requirements are listed in Table 1 below.

Table 1: Customer Requirements

Customer Requirements	
Durable	Efficient
Quick to power up	Meets pressure profile provided
Meets flow profile provided	Inexpensive
Recycles hydraulic fluid	Attractive appearance
Uses existing parts	Works in pre-existing hydraulic system
Easy to manufacture	Easy to repair
Simple user interface	Low weight
Easy to install	Compact size
Easy to control output pressure	

The USEPA stressed the need for an efficient supercharger that is quick to power up, and meets the provided pressure/flow profile (Appendix D) on an on-demand basis. The supercharger must function as part of a pre-existing hydraulic system that has been developed for use in UPS trucks, where hydraulic power takes over at idling. The need is for a system that is both cost-effective and able to withstand road conditions for the lifetime of the vehicle. In order to save on manufacturing costs, it is preferred that the system will use pre-existing parts and will be light weight. It is also important that the system can be accessed for repair needs, and is easy to install into vehicles.

The USEPA contacted the team prior to Design Review II to stress the need for a supercharger system that easily controls the output boost pressure. The team has investigated methods and mechanisms of control on the output, whose characteristics were weighed to determine the most efficient and cost-effective solution. The selection of an output control mechanism is contained in the Prototype and Final Design Description Sections.

The customer requirements provided by the USEPA were used in developing a design prototype, with the final design reflecting customer requirements not fully achieved with the prototype model.

III. ENGINEERING SPECIFICATIONS

Engineering specifications, which are the dimensional and performance requirements agreed upon with the customer, act as a contract for what the design team must accomplish. These were decided upon by analyzing the customer requirements and determining workable engineering specifications that achieve the desired performance characteristics. Thus for every customer requirement, there is one or more engineering requirements strongly corresponding to it.

In order to translate customer requirements into a specified technical description of what needs to be designed, a Quality Function Deployment (QFD) diagram was completed. The QFD, as seen in Appendix C, shows the correlation strength between all customer requirements and engineering specifications, and allows the team to rank the importance of both customer requirements and engineering specifications. The team developed the QFD by discussing each rating and coming to a consensus on what number to assign the relationship or importance weight. By developing the QFD as a team, we prevented individual biases from heavily skewing the results.

By benchmarking the existing hydracharger the USEPA has developed, we were able to get a pressure/flow profile that we are looking to match. However, we would like to improve upon its efficiency as well as design flaws such as the gravity drain of hydraulic fluid. In addition, we would like to reduce the size of the hydracharger unit in order to make it more feasible for installation in a vehicle. The basis for the design of both the Garrett® HydraCharger™ and USEPA hydracharger is that they shoot high-pressure hydraulic fluid to rotate turbines, which power the superchargers. We are looking to improve upon this design method by utilizing a hydraulic pump instead, with a mechanical coupling to transfer the power to the supercharger.

The relative importance of customer requirements was determined by rating the strength of their respective relationships to the engineering specifications. In addition, the team assigned a weight for the normalized importance to the customer for each customer requirement. The sum of the correlations between a customer requirement and the engineering specifications was multiplied by the normalized customer requirement importance to determine the overall requirement importance rating. These customer requirement ratings were then able to be ranked. This resulted in the highest ranked customer requirement being that we meet the pressure profile provided by the USEPA, followed by meeting the flow profile, achieving high efficiency, and being quick to power up. The least important customer requirement ranking was appearance.

We correlated all of the customer requirements in the QFD with the engineering specifications listed in Tables 2 and 3 below. They were all either generated by assignment of specific values, or by engineering analysis of the system. Specifically, the assigned pressure and flow curves were our main design drivers, and we utilized them in the creation of a spreadsheet containing many of the performance characteristics listed above for the supercharger, hydraulic motor, and the energy transfer system between the two, which we selected to be a belt drive system. The starting point in the order of quantifying our engineering specifications was the output pressure of the supercharger. From this, and knowing the ambient air pressure, we were able to calculate the compression ratio for the supercharger. By finding this ratio, we were then able to determine the ratio for the volume flow rate in and out of the supercharger. Next, from our customer-requested value for the flow rate out, we found the volume flow rate into the supercharger from the ambient. From this information, we were able to determine some possible components for our system. Then we were able to make further calculations based upon the published data on one such component, the Eaton MP45 Roots-type supercharger, which is a positive displacement compressor [9].

The first compressor curve we utilized was the relationship between the air flow rate into the supercharger and its RPM, in order to generate 5 psi of boost pressure, which is sufficiently accurate for our purposes because 5 psi is approximately 106.6% of the boost pressure we need to create. From the curve, we were able to linearly interpolate the necessary RPM for the correct pressure and previously calculated volume flow rate into the supercharger. Next, knowing this RPM of the supercharger, we were able to use linear interpolation again in order to find the horsepower required to drive the supercharger, which for our purposes will be the power supplied from the shaft connected to either a sheave or gear. Since the power is equal to the rotational speed multiplied by the shaft torque into the supercharger, we were then able to find this torque value. At this point, all of the performance characteristics of the supercharger were determined, at least for this particular unit in order to give us a good idea of the ranges we will be considering. These ranges may all be found in Table 2 below.

Once the performance of the supercharger was characterized, we were able to generate concept solutions for our belt/gear drive system. Our current plan is to use a belt system, so we estimated its efficiency in the transfer of power from the motor to the supercharger. This allowed us to find the necessary output of power from the hydraulic motor, and thereby allowed us to determine appropriate pairs of motor torque and rotational speed. Since they have an inverse relationship for a given value of power, we were able to increase the torque and decrease the RPM by varying the belt drive ratio. We assumed the sheave on the supercharger side to be one inch in diameter and then varied the sheave size on the motor side, on the order of 2.5 to 7 inches. These ratios were then further developed into our exact system specifications, which are reflected in Tables 2 and 3 below. These specifications came from our understanding of the relationships between the system components and their characteristic equations. All of this engineering analysis we have completed since the first design review has been aided by the equations and performance characteristics we have found in a number of new sources, mostly in engineering design related books [4,5,7,9,10,13,17,18,19].

This new knowledge has allowed us to correctly match and select components to use for our prototype, and to ideally use in our final design proposal. The engineering specifications listed in Table 2 below reflect those of the prototype design of the on-demand supercharger system, while the specifications listed in Table 3 below correspond to the final design. While many of the specifications are the same, some are different due to the difference in the components we selected, but the process for calculation of all of the quantities was nearly identical. A more detailed discussion of the associated calculations is contained in the section on engineering design parameter analysis later.

Table 2: Engineering Specifications for Prototype

Quantified Engineering Specifications	
Manufacturing and Development Cost $400, actual: $180.13	Max Boost Pressure at Supercharger Outlet 4.69-5 psi
Ambient Air Pressure at Supercharger Inlet 14.7 psi	Supercharger Max Flow Rate 113 scfm
Supercharger RPM 5320 RPM	Hydraulic Motor Operating Pressure Below 2000 psi
Material strength, stiffness, durability As high as possible	Leakage of air/hydraulic fluid None
Hydraulic Motor Torque Output 97.25 lb-in	Hydraulic motor RPM output 2787 RPM
Hydraulic Motor Power Output 4.30 HP	Hydraulic Motor Size height 9.685 inches
Hydraulic Motor Size length 9.73 inches	Hydraulic Motor Size width 5.20 inches
Engine to Supercharger, Hydraulic Motor to High Pressure Reservoir Interface	Supercharger Power In 3.96 HP
Supercharger Torque In 46.87 lb-in	Supercharger Efficiency 57%
Supercharger Power Out 2.26 HP	System Thermal Efficiency Assumed 100%
Supercharger Intake diameter 2 inches	Supercharger Outlet diameter 2 inches
System weight 120 lbs	System Pressure Output Control Valve at hydraulic motor inlet Supercharger Bypass Valve
Supercharger Size length 8.27 inches	Supercharger Size width 7.09 inches

Supercharger Size height 4.92 inches	Coupling Unit RPM/Torque Conversion 2.4 : 4.6 Belt Ratio
Coupling Unit length 2.5-3 inches	Coupling Unit width 16-17 inches
Coupling Unit height 9.8 inches	Belt System Efficiency 92%
Hydraulic Motor Efficiency 64%	Total System Efficiency 34%

The two most important engineering specifications listed above are the maximum boost pressure out of the supercharger and its maximum flow rate, both of which were obtained from the performance curves given to us by the USEPA. The on-demand supercharger prototype and final design will match all of the technical specifications that we have developed, with the aforementioned pressure and flow profiles being the main design drivers. It should be noted that the final design components differ from the prototype components, so some changes will occur, as can be seen below.

Table 3: Engineering Specifications for Final Design

Quantified Engineering Specifications	
Manufacturing and Development Cost $250,000 - $500,000	Max Boost Pressure at Supercharger Outlet 4.69-5 psi
Ambient Air Pressure at Supercharger Inlet 14.7 psi	Supercharger Max Flow Rate 113 scfm
Supercharger RPM 3800 RPM	Hydraulic Motor Operating Pressure Below 2000 psi
Material strength, stiffness, durability As high as possible	Leakage of air/hydraulic fluid None
Hydraulic Motor Torque Output 109.6 lb-in	Hydraulic motor RPM output 1990 RPM
Hydraulic Motor Power Output 3.46 HP	Hydraulic Motor Size height 6-9 inches
Hydraulic Motor Size length 7-10 inches	Hydraulic Motor Size width 4-7 inches
Engine to Supercharger, Hydraulic Motor to High Pressure Reservoir Interface	Supercharger Power In 3.18 HP
Supercharger Torque In 57.80 lb-in	Supercharger Efficiency 71%
Supercharger Power Out 2.26 HP	System Thermal Efficiency Assumed 100%
Supercharger Intake diameter 2 inches	Supercharger Outlet diameter 2 inches
System weight 100-120 lbs	System Pressure Output Control Valve at hydraulic motor inlet Supercharger Bypass Valve
Supercharger Size length 7-9 inches	Supercharger Size width 6-8 inches

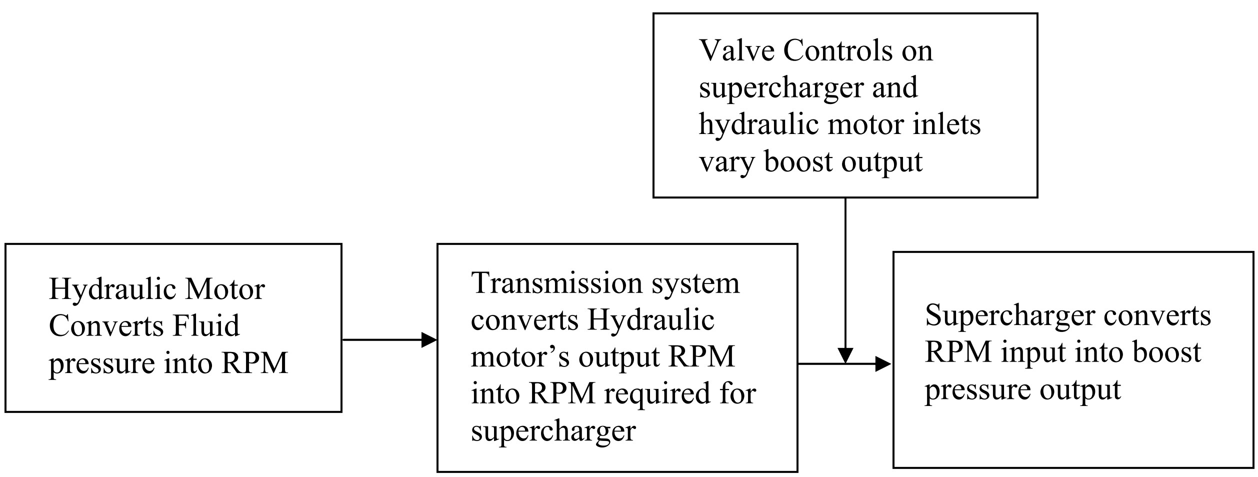
Supercharger Size height 4-6 inches	Coupling Unit RPM/Torque Conversion 1:1 Shaft Ratio
Coupling Unit length 2.5-3 inches	Coupling Unit width 2 inches
Coupling Unit height 9.8 inches	Belt System Efficiency 92%
Hydraulic Motor Efficiency 74.5%	Total System Efficiency 52%

IV. FUNCTIONAL DECOMPOSITION

To facilitate concept generation, a functional decomposition analysis was completed in a FAST diagram, which was expressly developed to decompose the sub-functions of the system. The FAST diagram, as seen in Appendix G, focuses on the sub-functions within the supercharger system, and allows one to generate concepts for each specific need. Specifically, the task function of the supercharger is producing boost. The basic functions of the supercharger that support the production of boost include transmitting power and pressuring the air. Finally, the primary supporting functions include increasing efficiency, simplifying operation, simplifying manufacturing, and performing adequately. After identifying these functions, the team expanded on the FAST diagram by determining how each function needed to be completed.

However, it is sometimes helpful to still be able to refer to the more generalized "black-box" functional decomposition for the on-demand boost supercharger can be described using a simple block diagram, as seen below.

Figure 1: Block Diagram of On-Demand Supercharger



The FAST diagram simply expands on this diagram by breaking down each of the components into their individual functions and sub-functions, which leads to our concept generation and selection process.

V. CONCEPT GENERATION & SELECTION PROCESS

Using the FAST diagram as a basis of what sub-functions exist within the supercharger, the team began developing concepts using the morphological chart. The morphological chart is a tool for brainstorming by which all sub-functions in the FAST diagram are listed in the left column, and corresponding concept solutions are drawn in the row to the right of each sub-function. The morphological chart can be seen in Appendix H. The concepts have been developed by designing for pressure, flow, cost, and efficiency.
The concepts for control of flow include varying incoming high pressure fluid flow with a valve which controls the variable displacement in the hydraulic motor, as well as varying the incoming air flow by changing the inlet diameter to the supercharger through the use of an attached butterfly valve. Both of these options could be used individually, or could be used in combination to achieve the desired system performance.

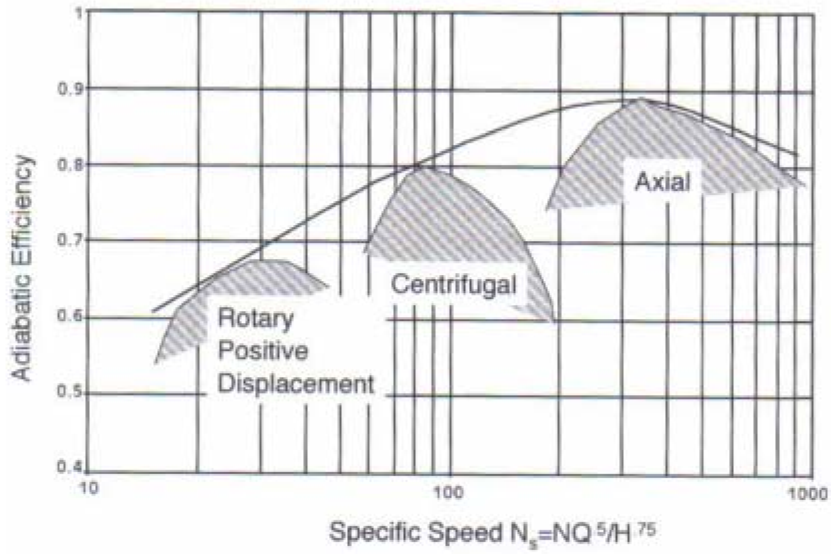
The user interface is also an important design aspect that the team considered. Possible solutions include a push-button or switch activation system on the steering wheel or shift-stick. However, for the purpose of testing, knob controls could be utilized to directly influence the output pressure of the supercharger.

The type of motor being used in the supercharger system is another design aspect. The team brainstormed the possible solutions of using a hydraulic motor, using a hydraulic generator attached to an electric motor, and using a hydracharger paddlewheel. A hydraulic motor would provide torque via a gear or belt system to the supercharger impeller. The hydraulic generator attached to an electric motor would store energy in the electric motor, decreasing the lag time associated with a direct connection between the supercharger and hydraulic motor. Finally, the hydracharger paddlewheel is the current design solution that the USEPA has developed. The team would develop a similar supercharging unit that uses compressed fluid energy and also has increased efficiency and cost-effectiveness.

The connection between the motor and supercharger must provide an efficient and durable transfer of energy. A gear system, belt system, or direct shaft connection are all design possibilities. The gear system would tend to be more sensitive to many cycles of use and more expensive than a belt system. A belt system would allow the user to adjust the sheave ratio more easily than adjusting the gear ratio. Finally, a direct shaft would be the most efficient connection, but would not allow for adjustment of energy transfer and would require a 1:1 ratio between the motor output and supercharger input. Based on the importance of simplicity and the short time frame in which this design must be implemented, the group has been exploring options with a belt system.

The choice of supercharger type is important due to the varying range of efficiency between positive displacement (screw), centrifugal, and axial types. Due to the low specific speed of our supercharger system, the positive displacement and centrifugal type casings are the most efficient for our application. The relative specific speeds and efficiencies for different supercharger types are shown in Figure 2 below [5].

Figure 2: Efficiencies and Specific Speeds of Supercharger Types



The axial type supercharger has the highest efficiency, but its fragile and expensive blades, extremely high-speed application and bulkiness puts it beyond the scope of this project. This knowledge of advantages and disadvantages was gained from Table 4 shown below [10].

Table 4: Relative Comparison of Compressors

Type	Advantages	Disadvantages
Centrifugal	Wide operating range Low maintenance High reliability	Unstable at low flow Moderate efficiency
Axial	High efficiency High-speed capability Higher flow for given size	Low pressure ratio per stage Narrow flow range Fragile and expensive blading
Positive displacement	Pressure ratio capability not affected by gas properties Good efficiencies at low specific speed	Limited capacity High weight-to-capacity ratio
Ejector	Simple design Inexpensive No moving parts High-pressure ratio	Low efficiency Requires high-pressure source

Based on the variety of possibilities for each component, we can produce a multitude of different combinations for our system. A detailed compilation of concept drawings can be seen in Appendix K. For instance, we can select a system powered by a hydraulic motor, which transfers power to a positive displacement supercharger by means of a gear system. It controlled by means of a valve regulating the pressure from the high-pressure reservoir. We can also use the same concept, except use a belt to drive the supercharger instead of gears. By varying individual components, hundreds of different concepts can be generated. The five most diverse combinations will be discussed below in the concept design process.

1. Hydraulic Motor-Centrifugal Compressor-Belt Driven-Pressure Valve Control
2. Hydraulic Generator/Electric Motor-Centrifugal Compressor-Gear Driven-Outlet Valve Control

3. Paddlewheel Hydracharger-Screw Type Compressor-Directly Driven-Pressure Valve Control
4. Hydraulic Motor-Screw Type Compressor-Gear Driven-Outlet Valve Control
5. Hydraulic Generator/Electric Motor-Centrifugal Compressor-Directly Driven-Pressure Valve Control

These five design variations are listed and rated according to the customer requirements in the Pugh chart, as seen in Appendix I. The design variations are rated according to whether they meet the respective customer requirement better, worse, or the same as the benchmark paddlewheel hydracharger. By creating a weighted score for each of these designs, we are able to evaluate the pros and cons of our options, thereby keeping the function aspect of our project in the forefront of the design and concept selection process.

We selected a design that is simple, cost effective, durable, and operational within our established boundaries. This document contains discussion of five possible designs that illustrate some of the design decisions involved. These five designs are detailed in Appendix I, which scores each design on its ability to meet customer requirements (listed in the QFD, Appendix C).

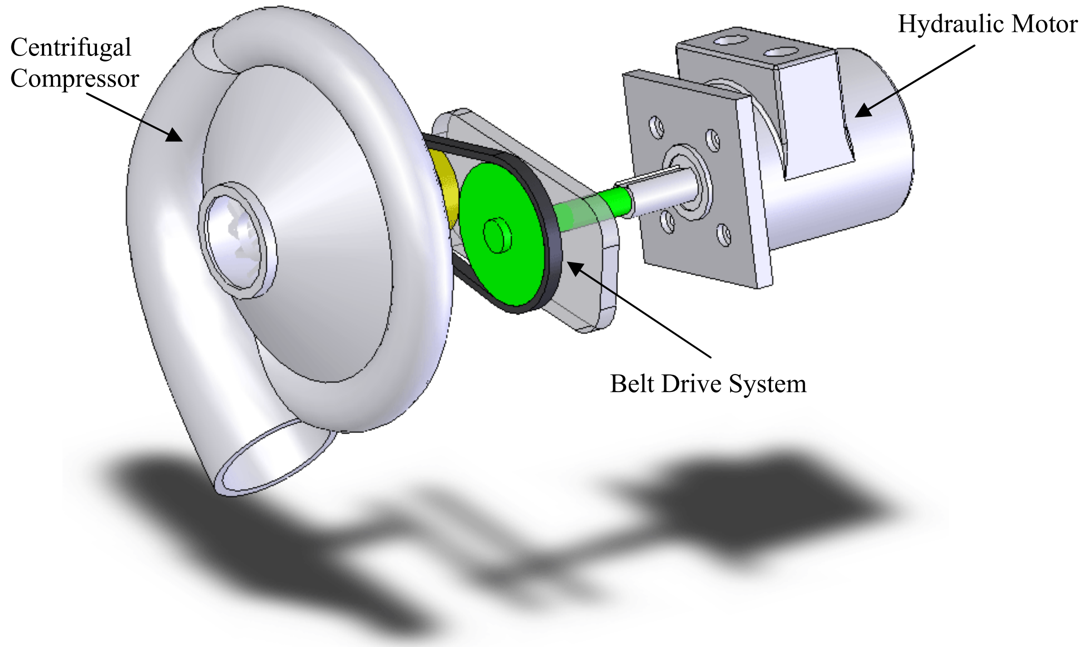
As seen in the Pugh chart, concepts B and C were ranked second to worst and worst, respectively, in terms of the five concepts meeting the customer requirements. Concept C involves the same design as the benchmark, except that the centrifugal supercharger is replaced by a screw type supercharger. This does not improve performance metrics such as overall system efficiency and ability to power up quickly. In addition, it increases both the weight and cost of the system based on the characteristics of the interchanged roots type supercharger. Concept B involves an electric motor attached to a hydraulic generator, which runs a gear driven transmission to a centrifugal compressor. The use of both an electric motor and hydraulic generator means that this concept is more expensive, heavier, and harder to install than the existing benchmark, and does not out-perform the benchmark in terms of performance metrics. In addition, gear drives tend to wear out, so durability is another concern. Concept D involves a screw type compressor, where the torque from a hydraulic motor is transmitted via a gear drive. Although this concept would improve efficiency and quickness to power up, its cost, weight, and durability are not improvements on the benchmark.

This leaves concepts A and E as the first and second ranked concepts. As a redundant measure of capability, the final two designs were then reassessed for scoring relative to each other. In this way, our group could directly correlate the advantages and disadvantages of each and pick the best option, reflected in our alpha design concept.

VI. ALPHA CONCEPT DESCRIPTION

The alpha design, as seen below in Figure 3 and in detail in Appendix K, represents the best known option for matching customer requirements as of Design Review 2, but has since been modified into two different forms, our final design and our prototype design. This initial alpha design was derived from the evaluation of performance for different components within the assembly using the Pugh chart in Appendix I. This chart allowed us to evaluate how well each concept meets customer requirements, and based upon the same method of analysis we have since found a better method of meeting the customer requirements, as can be seen in our final design.

Figure 3: Alpha Design Assembly



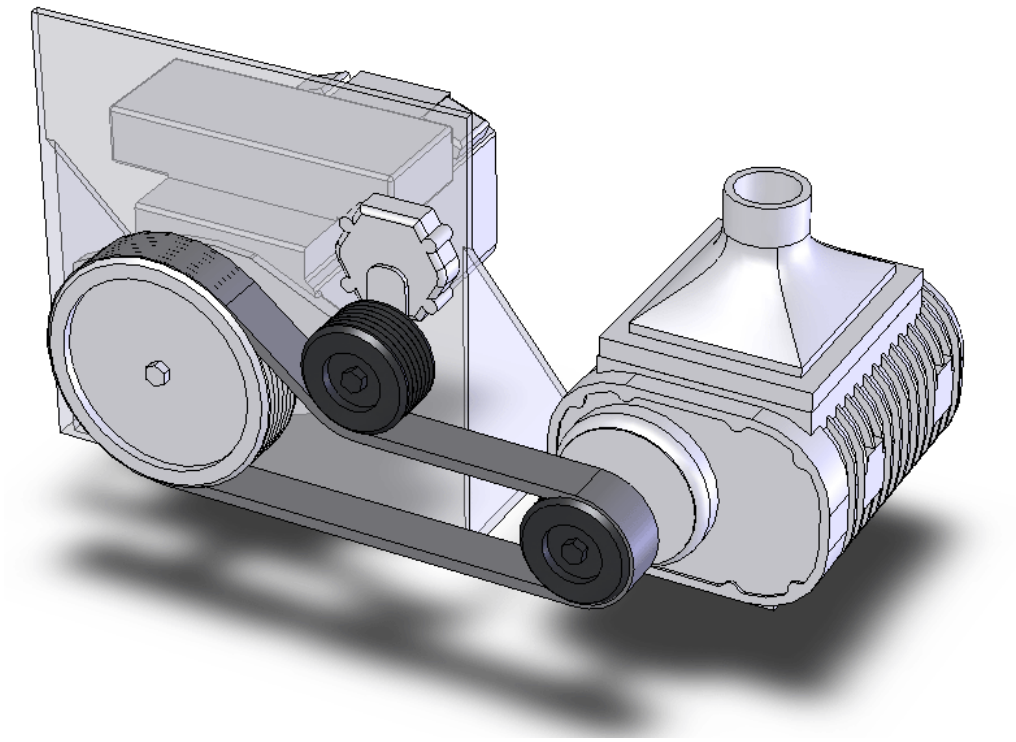
This design presents a versatile performance with regard to pressure output, efficiency, simplicity, and control. However, between Design Reviews 2 and 3 we found that there were better options for the final design as well as for the prototype design.

VII. PROTOTYPE/FINAL DESIGN CONCEPT DESCRIPTION

The prototype design for the on-demand supercharger system no longer reflects the basic form of the Alpha Design above, due to a number of design modifications. The first of these is the fact that a Roots-type supercharger will be used instead of a centrifugal compressor. Despite the advantages in efficiency cost for the centrifugal compressor, the team will use a Roots-type supercharger in the prototype after receiving one cost-free from the USEPA. This type of supercharger will be acceptable in the prototype, since the only real setback is a decrease in efficiency; therefore it still offers a full proof-of-concept validation. The second major change in the modification of the alpha design to the prototype design, as well as the final design, is that a system for tensioning the belt was created, and it went through multiple design iterations itself, as can be seen in the section on the final design description later.

The change from the alpha design above to the final design shown below incorporates the same ideas for the belt tensioning system, but includes a different type of supercharger which provides increased efficiency, with the only problem being that it does not fit the $400 prototype budget. Specifically, we include another variation on the screw-type supercharger called the twin-screw supercharger, which will be discussed further in the final design description. The picture above serves as a good representation of both the final design and prototype design since the only differences are in the shape of the helical lobed gears inside the superchargers and the specific model of the hydraulic motor.

Figure 4: Initial Prototype Design Assembly



The energy transmission between the hydraulic motor and supercharger could be accomplished via a belt drive, gear drive, or direct shaft connection. The advantages for belt drives include that they are used in application where the rotational speeds are relatively high, no lubrication is required, less noise, and they can be used for long center distances. Gear drives are more compact than belt drives, and have greater speed capabilities. Metal gears do not deteriorate much with age, heat, or grease.

A belt/pulley system will be used in the final design due to the ability to easily adjust the transmission ratio. A grooved belt will be used and will ride on the sheave of the hydraulic motor with mating grooves. However, in using a belt drive, one must worry about the high tension in the belt at extremely low speeds, as well as the e, while at high speeds centrifugal forces, belt whip, and vibration all contribute to belt life decreasing.

VIII. ENGINEERING DESIGN PARAMETER ANALYSIS

Determination of all of the engineering specifications was completed in a number of steps. First we needed to do a thorough analysis of the customer requirements provided, specifically the pressure and flow targets. Using these targets as our design drivers, from this information we were able to begin the creation of a spreadsheet detailing all of our engineering specifications related to the performance of the system. First, we entered all of the fundamental data related to the fluid mechanics and thermodynamics involved with the operation of the supercharger. We assume that the air intake of the supercharger is at the ambient air pressure, 101 kPa. We also use quantities associated with standard air temperature and pressure. Specifically, we obtained values for the air's density (ρ), specific weight (γ), dynamic viscosity (μ), kinematic viscosity (ν), gas constant (R), and specific heat ratio (k). Here we will consider all of these as knowns, and all of these values may be seen in the section containing a more detailed development of the equations relating to engineering analysis, in Appendix R. We will be considering the case of the supercharger to be an adiabatic, isentropic process, which results in $PV^n = PV^k$, therefore n is equal to k, which has a value of 1.4, which may be utilized in the next steps.

The development of our exact specifications requires multiple steps in that we needed to know what might be possible in an ideal system first. Then from that point, using appropriate components and their associated performance characteristics, we needed to find out how a real system will perform, specifically we need to characterize the performance of our prototype system in order to make a valid comparison.

The first step is to analyze the performance of an ideal system. Ideal or not, the system we are creating must satisfy the engineering specifications we have set out previously in this report. Specifically, we need the supercharging unit to take the ambient air pressure, 101 kPa, and add to this the additional pressure we want to create at the output of the supercharger, 33 kPa, resulting in an absolute output pressure of 134 kPa. From this information we were able to move forward in our calculations, first by looking at the ratio of output to input pressure.

$$R_P = \left(\frac{P_{o,absolute}}{P_{i,absolute}} \right)_{SC} \approx 1.327 \qquad (1)$$

This equation then allowed for calculation of the ratios of the output to input temperature and volume based on the value of n=1.4, as discussed above.

$$R_V = R_P^{(1/n)} \approx 1.224 \qquad (2) \qquad\qquad R_T = R_V^{(n-1)} \approx 1.084 \qquad (3)$$

Knowing these ratios and knowing the values for the input density and temperature, we are then able to find these quantities for the output. However, these facts are not critical here so they are left to Appendix R. What is important is to take into account not only the pressure rise we need to create, but also the desired volume flow rate. We were provided with the figure for the input and output standard flow rate Q=113 standard cubic feet per minute (scfm). We assume that the inlet of the supercharger is at standard air temperature and pressure, so the inlet volume flow rate in actual cubic feet per minute (acfm) will be equal to the 113 scfm given. However, we know that the outlet will be pressurized and will be at a higher temperature. Thus the supercharger outlet and will therefore have a different value for its flow rate in acfm, which we calculated to be about 137.7 acfm based on Equation 2 above.

$$Q_o = \left(Q_i \times R_V\right)_{SC} \approx 137.7 \; acfm \qquad (4)$$

For use in further calculations, both the input and output flow rates were converted to a number of different sets of units. The inlet and outlet air velocities, as well as the mass flow rate of air through the supercharger may also be calculated at this point by dividing the volume flow rate by the cross-sectional area of the inlet and outlet, but this information is not necessary for the evaluation of the idealized performance of the system. Even in the case of the real system, as long as the inlet and outlet have a sufficient area, there should be virtually no losses since there will be no clogging of the air as there would be in the event that the area was too small, as could be imagined in the limiting case that the outlet was infinitely small.

The next step in the necessary analysis is to find the adiabatic head H_{ad} produced by the supercharger in pressurizing the air, which requires calculation of the quantity ZRT as shown below in equation 5.

$$ZRT = P_{o,\,abs} \times v_o \approx 89477\ J/kg \tag{5}$$

This quantity is needed to use equation 6 to find H_{ad}.

$$H_{ad} = ZRT \times \frac{k}{k-1} \times \left(R_p^{\,(k-1)/k} - 1 \right) \approx 25785\, m^2/s^2 \tag{6}$$

This value for the adiabatic head is needed to find the minimum amount of power that could possibly be provided in order to supply the required pressure and flow rate for the supercharger. This ideal minimum power needed can be calculated by equation 7 below.

$$PWR = \left(\frac{H_{ad} \times Q_i \times 144 \times P_{i,\,abs}}{\eta_{ad} \times T_{abs} \times R_{air} \times 33000} \right) \approx 2.260\ HP \tag{7}$$

This can be used later in the evaluation of the efficiency of the prototype and final design in that the actual power required will be greater than this minimum power.
The first step in the development of our prototype's exact theoretical specifications was to choose an appropriate supercharger and obtain its performance curves. We accomplished this with the Eaton MP45 model supercharger, for which we had plots of its inlet flow rate vs. speed rpm, power hp vs. speed, and delta T vs. speed. Each of these plots contains a curve corresponding to a boost pressure of 5 psi, which is 106.6% of our boost pressure goal of 4.689 psi, so this should be a good estimation of what we need to create, while giving us a little room for improvement.

With this knowledge, we were able to move forward and use linear interpolation on the performance plots to find the desired characteristics. First we used the previously provided inlet volume flow rate of 113 scfm to find the corresponding value for the supercharger's speed, about 5320 RPM. Knowing this, we were able to use linear interpolation on the next curve to find the corresponding power needed to be provided to drive the supercharger at the desired speed and pressure, which yielded a value of 3.956 horsepower (HP). Knowing this power, we were able to begin to move towards the calculation of the requirements for our power transfer and power supply systems.

With values known for the rotational speed of the supercharger and the power needed to be supplied, this allows for calculation of the torque needed to drive it as we desire. We used a conversion based on equation 8 below to find a value of about 3.906 ft-lb.

$$\mathrm{T}_{in,SC} = \left(\frac{P_{in}}{\omega} \right)_{SC} \approx 3.906\ ft-lb \tag{8}$$

With the torque into and rotational speed of the supercharger known, all that remains to be done is to select an appropriate hydraulic motor and design the power transfer system based on its performance characteristics, specifically its maximum efficiency point. Based on our power requirement for the input to the supercharger, we used a belt efficiency of 92% to calculate the hydraulic motor power output requirement to be about $P_{out,\,HM}$ = 4.245 HP. Based on this power output and the speed ranges we are working with, we selected a motor with a maximum efficiency of about 88%. This efficiency should be achieved in the middle of its RPM range, since efficiency will be lost at either extreme. The motor is rated for 5550 RPM, therefore we will want to operate at or near ω_{HM} = 2775 RPM. This allows for a calculation of the drive ratio by equation 9 below, as well as calculation of the torque output from the hydraulic motor by equation 10 below.

$$R_{Drive} = \left(\frac{\omega_{SC}}{\omega_{HM}} \right) \approx 1.909 \qquad (9) \qquad\qquad \mathrm{T}_{out,\,HM} = \left(\frac{P_{out}}{\omega} \right)_{HM} \approx 8.001\ ft - lb \qquad (10)$$

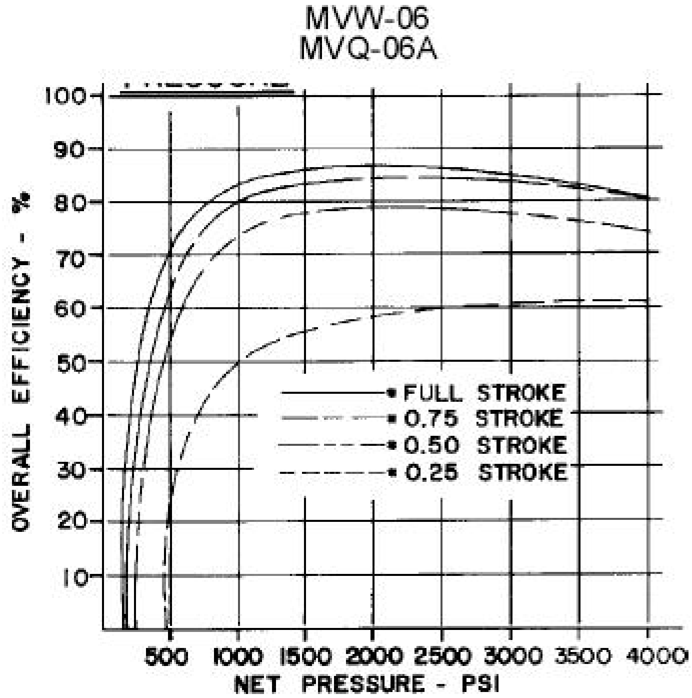
Knowing the desired drive ratio, we must decide on dimensions for the power transfer system, which we chose to be a belt drive system due to its existence on the supercharger we selected. This supercharger already has a pulley attached to it with a 2.4-inch diameter. Therefore, multiplying by the drive ratio, we find that the pulley attached to the hydraulic motor needs to have a diameter of about 4.6 inches. Any variation from this will simply make a slight change in the motor's speed and torque output that we will need to be aware of during the operation of the supercharger.

Now that we have both the ideal system and actual prototype system characterized, we are able to analyze its performance in terms of efficiency. We find based on equation 11 below that the efficiency of the supercharger unit in our prototype will have an efficiency of about 57.1%.

$$\eta_{SC} = \left(\frac{PWR_{\min}}{PWR_{actual}} \right)_{SC} \times 100\% \approx 57.1\% \qquad (11)$$

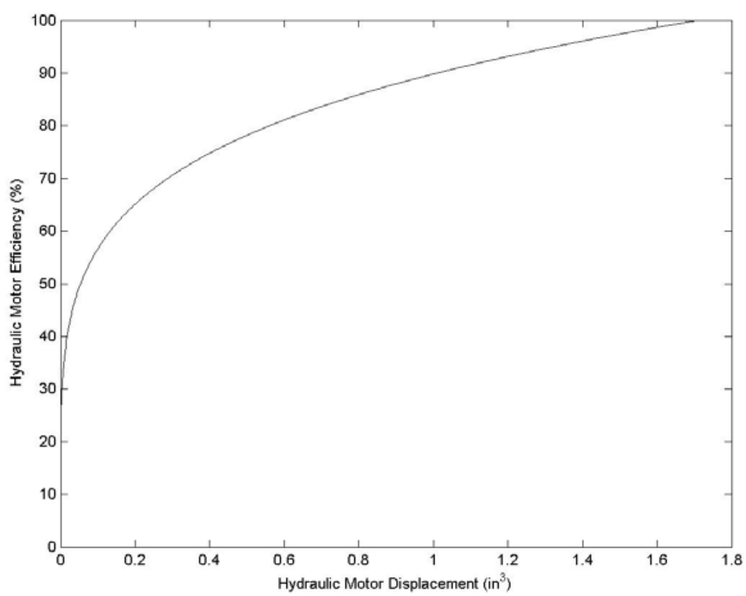
To find the total system efficiency we also needed to know the efficiency of the hydraulic motor. We used performance curves from the Oilgear variable-displacement hydraulic pump to gain an understanding of the general behavior of the efficiency of a hydraulic motor with regard to how it changes as the displacement changes to various fractions of the full available displacement. These curves, as can be seen below in Figure 5, show that as the displacement decreases, so does the efficiency, but not in a linear manner. As a result, due to the fact that we do not have published efficiency curves for the Bosch-Rexroth hydraulic motor we are used in our prototype, we must use this understanding of the nonlinear efficiency-displacement relationship to make our best possible engineering judgment as to the efficiency of the motor and as a result, the entire system. The curve below also illustrates a critical aspect of the decision-making process that went into the sizing of the two pulleys, which is that the best efficiency is generally located near the middle of the range of speeds of the motor.

Figure 5: Oilgear Hydraulic Pump Performance



From this curve, and knowing the range of operating pressures we will be dealing with, we were able to calculate the average displacement we will use in the hydraulic motor, which led to an estimation of the hydraulic motor efficiency, which will always be a function of displacement. Specifically, with the knowledge of the maximum motor efficiency and of the way that this efficiency decays, we were able to generate what we believe to be a good representative relationship between the two, which can be seen in Figure 6 below.

Figure 6: Representative Hydraulic Motor Efficiency-Displacement Relationship



From the figure above, we can find from numerical testing done later in this report that the steady-state average hydraulic motor efficiency will be about 67%. This relatively low result is due to the fact that we will be using on the order of 15%-30% of the available displacement in the hydraulic motor from our prototype design. Then multiplying this motor efficiency by the belt efficiency of 92% and the supercharger efficiency found earlier as 57.1%, we reach an overall minimum system steady-state efficiency of about 33.6% for our prototype design, as a minimum. However, it should be noted that during the start-up phase, because more of the available power from the hydraulic motor can be utilized, the efficiency during this phase will be much higher than in the steady-state case, due simply to the fact that the motor will be at its maximum efficiency. This difference will be discussed further in the Validation and Simulation Results section later in this report.

In all phases, both start-up and steady-state, this system efficiency of the prototype can be improved in the final design by a few factors. One possibility that we do not explicitly discuss, but should be considered, is that direct coupling of the hydraulic motor and supercharger could eliminate the need for a belt system. A simple replacement is made by the fact that we selected a Lysholm twin-screw supercharger which would operate around 71% efficiency based on the performance curves found in Appendix S. It would also be improved by using a custom-designed hydraulic motor with a smaller, precisely selected maximum displacement, which will allow for more efficient operation, likely near or even above 80%. Using the same belt efficiency as before, 92%, we arrive then at a total system efficiency of around 52%. Thus the prototype is not too significant a setback from the final design, since it creates a total efficiency decrease of about 18%, and it can certainly be used as a proof-of-concept. In addition, the prototype should be very helpful in determining the parameters necessary for the design of the custom hydraulic motor, specifically the maximum displacement desired. By designing a motor with a smaller maximum displacement, the efficiency during steady-state operation will be improved. However, this should only be done to a certain extent, since it is helpful to have more available displacement during start-up for a faster speed-up time. Again, this will be discussed further in the Validation and Simulation Results section later.

Other engineering specifications for both the final and prototype designs, such as material selection, have been completed by a simple logic process. In order to ensure that the hydraulic motor is constructed with sufficiently strong materials, we made sure that the component we selected for the prototype, the Bosch-Rexroth AA6VM variable-displacement 28 cc hydraulic motor, was rated for the full 5000 psi of hydraulic pressure that it may be supplied with. The same idea also applies to the selection of the Eaton MP45 supercharger unit, in that the pressures, flow rates, and RPM we will be using are within its specified limits, and in fact we will be operating this unit in the lower range of what it is capable of handling.

Some of the critical loaded parts which we will manufacture ourselves are the sheaves for the belt drive system and the mounts for this system. In order to minimize the risk of any pieces fracturing or fatiguing, we completed some elementary engineering analysis of the loads these pieces will see, and accounted for this by deciding on the appropriate part dimensions accordingly. In the case of the bending load put on the shaft connecting the hydraulic motor to the pulley, we need to account for the tension put in the belt. Here we need to worry not only about the strength of the shaft, but also the belt itself, as we do not want the belt to slip excessively, causing undue wear. To prevent this problem, we selected an appropriate belt based on its cross section and operational limit relative to our operating speeds.

For additional parts that we will be manufacturing ourselves, such as the mount for the supercharger system, belt tensioner, supercharger outlet manifold, and pulleys, the materials chosen are aluminum and steel. This is based on their tensile strength properties, low cost, low weight, and their ease of machining. Finite Element Analysis (FEA) was not necessary in verifying the suitability of using these materials, as the forces and moments applied are for the most part absorbed in the steel test stand.

The design is being modeled in component form with separate analysis of the supercharger system, the hydraulic motor system, and the energy transfer system. Then, these three sub-systems are unified to analyze the performance of the system as a whole. The level of analysis is appropriate in that all major design factors were taken into account, while auxiliary factors such as the rotational inertia induced losses in the energy transfer system were assumed negligible. These auxiliary factors did not affect the modeling of the system or the concept development process. We have confidence that our analysis is correct because we have taken into account all factors which we consider to greatly influence the performance of the supercharger. The analysis relates to our physical prototype in terms of the component selection process and energy transfer system characterization. The one place where we have made an engineering approximation is in the performance of the hydraulic motor. This is because we were unable to obtain the performance characterization curves for the motor we will be using in the prototype, only for the hydraulic motor in the final design. These engineering approximations are detailed in Appendix R. If we are able to obtain these performance curves, further analysis will be conducted to determine definitive characterization of the prototype system.

Design for Manufacturing and Assembly

In addition, a Design for Manufacturing and Assembly (DFM&A) document was developed in order to ensure that the components are easy to manufacture as well as assemble. Since the design engineer casts the largest shadow in determining product cost, it is important to identify high-cost production processes and make design changes to reduce these costs. Within the DFM&A, as seen in Appendix P, each component is ranked for categories relating to manufacturability and ease of assembly. The belt/pulley system resulted in the largest total weighted ranking of all parts. This reflects the need to improve the ease of manufacturability for the belt/pulley, as well as making design changes to improve assembly time.

As shown in Appendix Q, design changes are shown reflecting the need to have a design that is easy to manufacture and assemble.

Design Failure Mode and Effect Analysis

A Design Failure Mode and Effect Analysis (DFMEA) was developed for the on-demand supercharger system in order to identify potential failures, determine the effects of these failures, and undertake preventative actions to avoid such failures. The DFMEA is important in assuring the safety of a product, and is important in how it documents engineering changes related to failure modes. It also brings to the attention of design engineers certain failure modes that can allow them to make design changes early on in the timeline of a project. Changes in design late in a project's timeline can be very costly.

As the supercharger and hydraulic motor were acquired through the US EPA, and did not involve the team directly manufacturing their individual components, the team created a system DFMEA to focus on the interaction between components, as seen in Appendix O. Weights for severity, occurrence, and detection were assigned for each failure mode, and a corresponding risk priority number (RPN) that shows the design team the relative urgency of design changes to eliminate failure modes.

With an awareness for all failure modes maintained, the design team highlighted failure modes with RPN's greater than 100 as urgent for design change and/or detection implementation. The belt/pulley assembly has 12 of its 14 failure modes corresponding to critical RPN values, with user interface accounting for one highlighted failure mode.

Possible modes of failure for the belt/pulley system include the belt material failure of the belt due to incorrect selection of belt material, excessive tension, excessive heat, debris in pulley grooves, or drive pulley misalignment. Any yielding of the belt would ultimately lead to a complete loss of boost pressure. In addition, the leaking of hydraulic fluid onto the belt will reduce the frictional characteristics and lead to loss of torque through slippage, which would decrease the boost pressure of the supercharger. The shaft of the pulley must also withstand the high rotational speeds needed while efficiently rotating within the bearing. Possible failure modes for the pulley include shaft material failure and friction in the shaft due to improper installation or incorrect surface finish and bearing lubrication. As a system, the grooved belt could jump its alignment with pulley grooves, leading to a complete loss of boost pressure.

For an example of a failure mode with an RPN value that can be largely ignored, the failure of objectionable squeaks or vibrations has an RPN of 3. This is because the severity of noise in the scope of other failures of the system is low, and will not cause system failure or malfunction. It is merely an example of a failure mode that the customer might be dissatisfied but doesn't affect the performance of the product.

IX. PROTOTYPE DESIGN DESCRIPTION

While the final design of our project would be the most optimal overall proposal provided we have the necessary funding and lead time, the actual prototype that will be presented at the design expo will be built with these limitations taken into consideration. The primary affect of these limitations on our project is a reduction in our overall maximum system efficiency of about 15%. Aside from the disparity in efficiency however, the prototype will still be a similar variation of the final design in terms of scale, functionality, performance, cost, and engineering specifications.

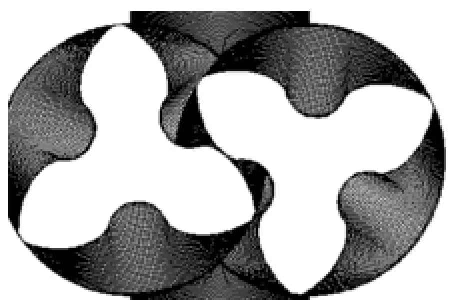
The prototype proves the most important elements of the final design by allowing us to determine whether the pressure and flow profile can be achieved with the given hydraulic motor and compressor. Through validation testing, we will be able to evaluate the feasibility and performance of the final design. The list of prototype components is shown in Table 5 below.

Table 5: Prototype Design Components

Component	Function	Justification	Cost
Eaton MP45 Supercharger	Pressurize air at intake Increase air flowrate	Meets requirements for output Air pressure 4.69 psi Flow 113 SCFM	$2900
2.4", 4.6" Pulley	Transfer RPM, Torque to supercharger from hydraulic motor	Pulley diameter based on derivation of RPM from Supercharger/hydraulic motor operational capability Material: Aluminum	In house (~$100)
Belt Tensioner	Tighten Belt	Simple, effect, standard means of tightening belt Material: Aluminum	$50
Bosch-Rexroth AA6VM Hydraulic Motor	Spin belt drive to accommodate appropriate supercharger RPM	Meets requirements for RPM Matches calculated requirement for torque Can be adjusted with ΔP	$5000
Supercharger Manifold	Provides outlet from supercharger into engine manifold	Meets requirements for output Material: Aluminum	In house (~$50)
Supercharger system mount	Provide mounting surface for supercharger system	Meets required size Material: Aluminum	In house (~$200)

The total cost of the prototype design is $8300. An illustration of the operation of the Eaton MP45 supercharger is seen in Figure 7 below.

Figure 7: Air flow inside Eaton MP45 Roots-type supercharger



Since the majority of our system components are high-performance machinery, specifically the supercharger and hydraulic motor, our budget of $400 is not nearly sufficient enough to purchase these parts from a manufacturer or supplier. Based on our final design, we would use a Lysholm Twin-Screw Supercharger, which operates with a volumetric efficiency of up to 80%. These types of superchargers typically cost over $4000. Due to this high cost, we decided to use a supercharger that we could procure from the USEPA which they already owned, the Eaton MP45. From our calculations, it was determined that our selected prototype hydraulic motor only needs to provide around 4.3 HP to power the Eaton MP45 supercharger, which will operate at an efficiency of about 57%.

The Eaton Model P45 supercharger was chosen based on its ability to produce 4.69-5 psi of boost pressure with a flow rate of 113 scfm. The RPM operating range between 4000-6000 RPM met the needs for hydraulic pumps appropriate to the application. It pumps 0.75 Liters of air per revolution, and is designed for use with 2.0L to 3.0L passenger car and light truck engines. Thus it is appropriate for our application, at least as far as the prototype goes.

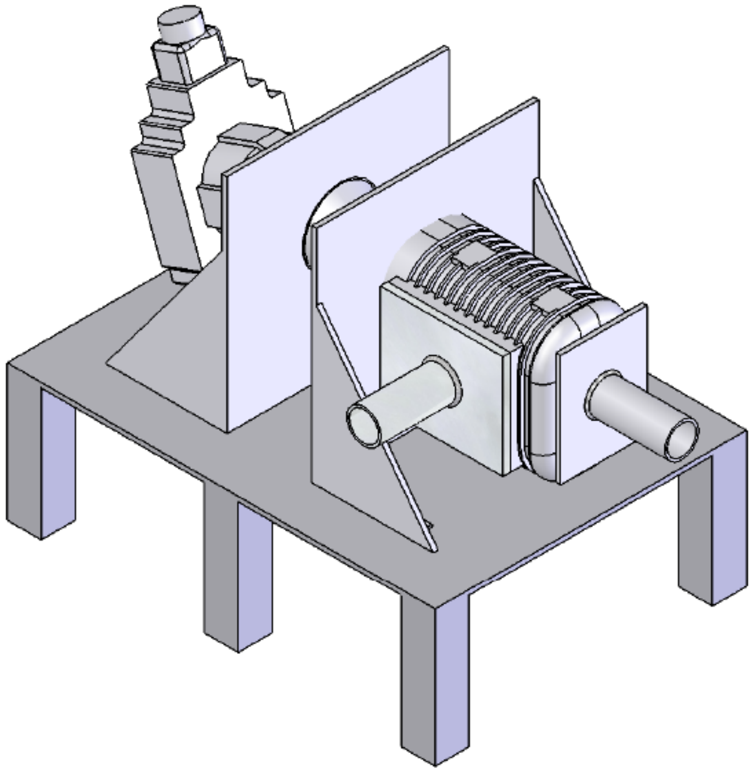
To power this Eaton supercharger at 5320 RPM, our desired speed, we needed a hydraulic motor that fit a reasonable range of sheave sizes, and we also wanted to use one that the USEPA already owned. We satisfied these conditions with the Bosch-Rexroth AA6VM 28 cc variable displacement hydraulic motor. With this motor, we are able to operate in the middle of its RPM range where it is most efficient, and can use a reasonable set of sheave sizes, 2.4 and 4.6 inches in diameter. The only problem is that due to its high power rating of 138 HP, we will have to use only a small portion of its available displacement, on the order of 15-30%, so this will decrease its average efficiency to about 67%, but yet again this is still an improvement over the fixed displacement motor that provides an average efficiency around 62%. Thus our prototype should perform in a reasonably efficient manner.

X. FINAL DESIGN DESCRIPTION

After several design iterations based on our research, engineering analysis, assessment of component capabilities, and the specifications from our sponsor, our team has developed a final design for our system. The purpose of this section is to present the final design and the individual components that make up the system.

The final design of our system is presented in Figure 8 below. It is composed of three major components: a Lysholm Twin-Screw 1200AX screw-type supercharger, a custom variable displacement hydraulic motor, and a custom spline coupling. The supercharger and hydraulic motor our positioned facing each other by means of mounting brackets and connected directly together by the custom coupling. Note that the supercharger is mounted on its side in order to allow easier flow into and out of the system. This positioning also reduces the over system height. For more detailed dimensional drawings of individual components and a bill of materials for our final design please refer to Appendices T and N.

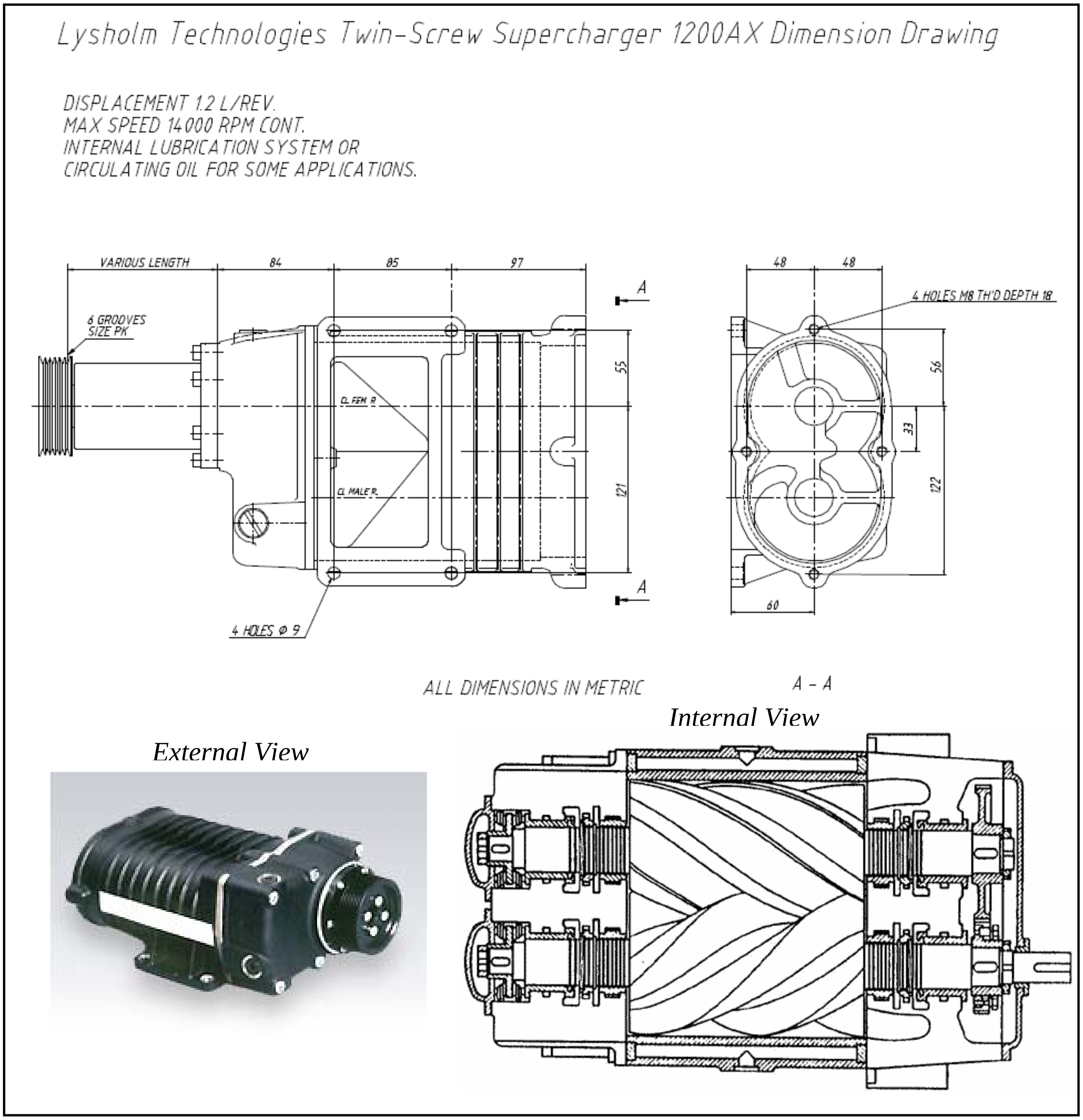
Figure 8: CAD Drawing of Final Design Concept



Lysholm Twin-Screw Supercharger 1200AX

Due to its relatively high efficiency at the RPM ranges that we are looking to operate at, the Lysholm Twin-Screw Supercharger 1200AX is our supercharger selection for our final design. The dimensional drawing, internal and external views of this model are displayed in Figure 9.

Figure 9: Dimension Drawing and External/Internal Views of the Lysholm Twin-Screw Supercharger 1200AX

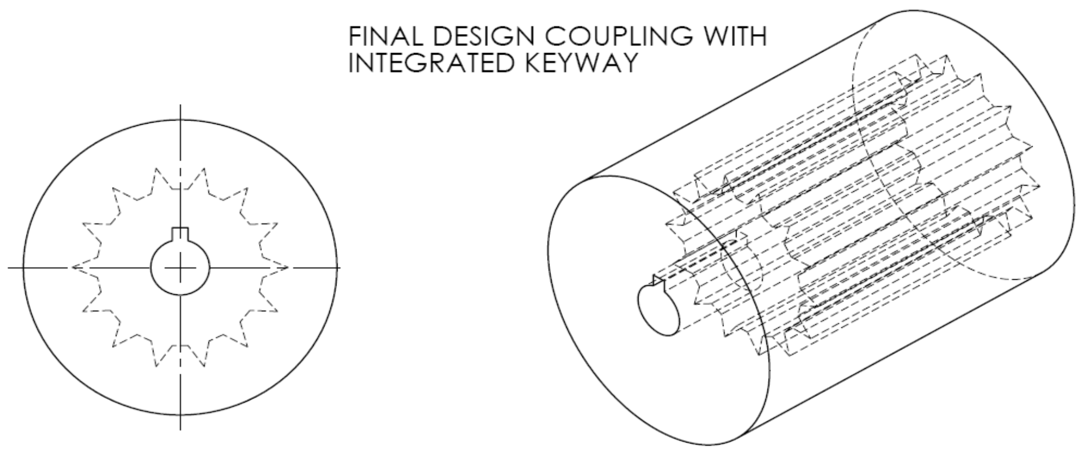


Custom Variable Displacement Hydraulic Motor

Even after a profuse amount of research, our team was unable to find a hydraulic motor that could efficiently produce the required power output with the robustness to operate safely at our high pressure range. Due to the significant efficiency loss that would result from using any existing variable displacement hydraulic motor, our team decided that it would be necessary to design and manufacture a custom variable displacement hydraulic motor. This motor would be designed based on the testing of our prototype in order to determine the value of maximum displacement that would ensure a high operating efficiency (at our desired power output), a fast start-up time, and have a steady-state output of our specified supercharger speed, all while being able to operate safely within our pressure range.

Custom Spline Coupling
Since our custom hydraulic motor will be operating at our specified supercharger speed, there is no need for a belt or gear system. Instead, a direct coupling of the hydraulic motor and our supercharger will be used, which will maximize our system efficiency and reduce system components and complexity. In order to couple the output shaft of our custom hydraulic motor and the drive shaft of the Lysholm Twin-Screw Supercharger 1200AX a custom spline coupling will have to be designed and manufactured. Based on the output shaft of our prototype's hydraulic motor and the drive shaft of the Lysholm Twin-Screw Supercharger 1200AX, the final design of the spline coupling is illustrated in Figure 10.

Figure 10: Final Design of Custom Spline Coupling



Prototype Validity of Final Design
Although our prototype does not use any of the components of our final design, it is a full-scale working model that will serve as a proof of concept and a means of testing for the final design. The prototype also validates the feasibility of manufacturing and assembly processes necessary for the final design and the geometry and material of the support structures. What the prototype does not validate is the performance characteristics of the system, the upper limit of system efficiency (due to the belt system and types of components used), and the selection of any of the components for the final design. Essentially, the purpose of the prototype is to serve as a validation for the continued research and development of the final design based on test results from the prototype.

Operation of Final Design versus Prototype
In terms of operation, the prototype design and final design are fundamentally the same. Figures X and X shows the transfer of hydraulic fluid, air, and energy for both the prototype and the final design.

Figure 11: Operation of Prototype Design

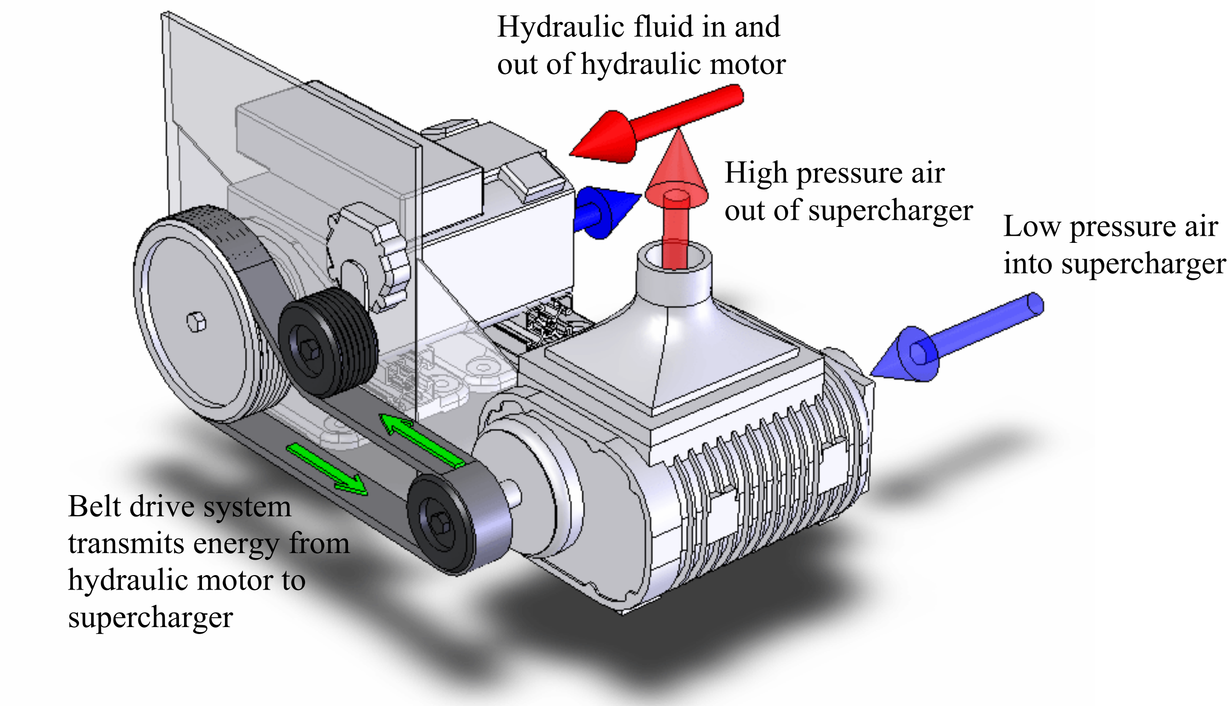
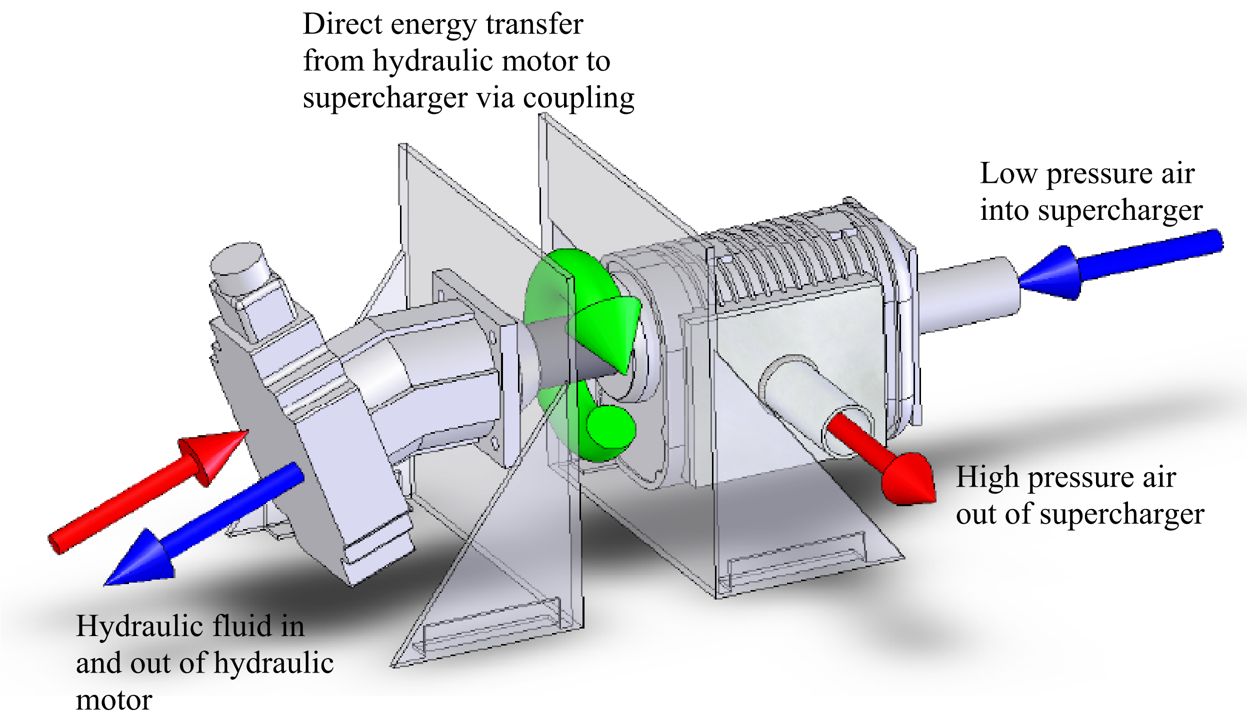


Figure 12: Operation of Final Design



The major difference between the prototype and final design in terms of operation is the method of energy transfer between the hydraulic motor and the supercharger. As mentioned above, the custom hydraulic motor eliminates the need of a belt or gear ratio to insure the proper operating speed of the supercharger; instead the final design employs a coupling to directly transfer the energy from the hydraulic motor to the supercharger.

Prototype Expectations

We expect our prototype to meet and possibly exceed the requirements outlined by our sponsor except for the efficiency of the system. For a detailed analysis of our prototype's validation and testing protocol, see section X below.

XI. MANUFACTURING PLAN

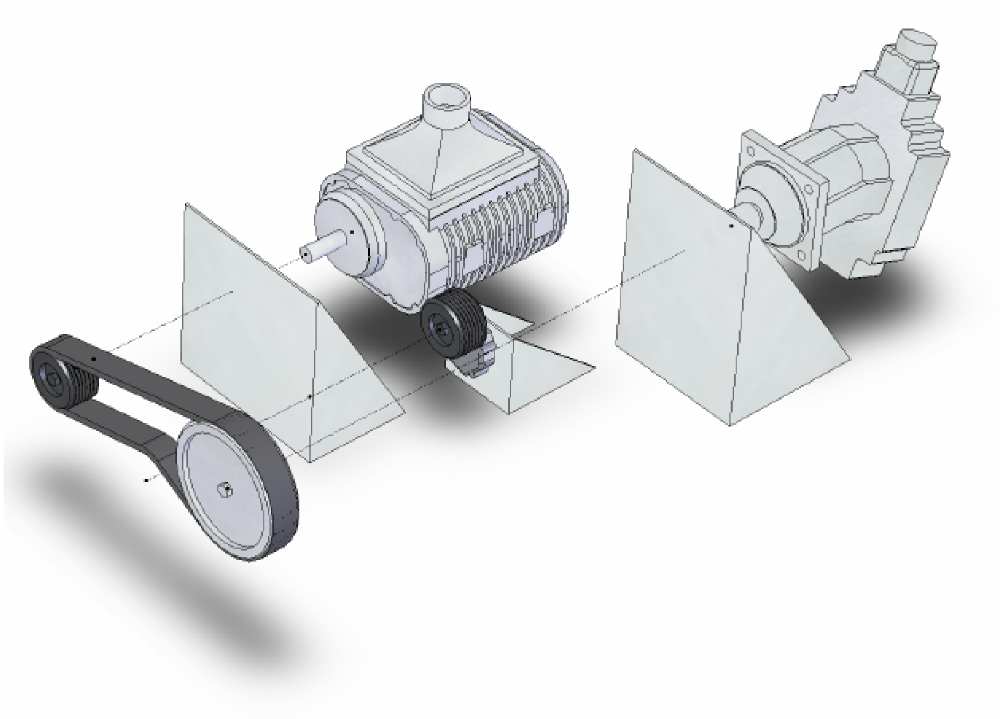
The fabrication of the prototype and final designs is centered on four part sections: the supercharger, the hydraulic motor, the test stand, and support hardware. Each of these aspects of the build will be focused on in the manufacturing plan. In general, two materials were used in the project, steel and aluminum. An outline of the various speed settings used on different machines is listed in Table 6.

Table 6: Machine process spindle and cutter speed rates for aluminum and steel

		Material, Speed	
Machine	**Unit**	Aluminum	Steel
Mill-end mill	RPM	1100	600
Mill-drill	RPM	400	80
Drill Press	RPM	400	100
Band Saw	FPM	290	100
Lathe	FPM	80	

As an overview, we have included an exploded view of our prototype and final designs in Figures 13 and 14. This figure depicts the designs with all of the individual components arranged linearly from their point of interface.

Figure 13: Exploded View of Prototype Design



Supercharger:
Manifolds

1. Use band saw - roughly cut 0.25″ 6061-T6 AL block to fit inlet and outlet of supercharger.
2. Square off edges of plates using 0.375″ end mill.
3. Mill 1.75″ hole at center of each plate- keep plate in mill vice.
4. Drill mounting holes for intake and outlet based on mounting holes specified for the supercharger in Appendix L.
5. Use band saw- cut two 4″ long, 2″OD tube, 0.125" wall thickness 6061-T6 AL, smooth the cut edge with hand file
6. Weld tube to centered plate

Bracket Mount

1. Use band saw - 0.125″ Steel plate to roughly 8.5″ square
2. Square off edges of plate using 0.5″ end mill.
3. Mill 2.6″ clearance hole centered at 6.1″ from the short edge of the plate using 0.5″ end mill- keep plate in mill vice.
4. Drill mounting holes for the supercharger as per Appendix L for the shaft side of the supercharger.
5. Remove plate from mill
6. Using new piece of steel cut two 8.5″ × 5.25″ triangles on the band saw.
7. Square off edges of plates using 0.5″ end mill.
8. Cut two 4″ long 90° angle steel using the band saw
9. Drill three 5/16″ clearance holes with the center hole at the center of one of the sides. 1″ spacing for the other two holes.
10. Clean all surfaces using acetone

11. Weld triangles along their long edge to the Steel plate with the majority of supercharger holes oriented at the top of the plate(narrow side of the triangles)
12. Weld angle steel to the inside of each of the 5.25″ bases of the triangles at 1″ from narrow edge of the base, holes exposed to the middle of the assembly.

Hydraulic Motor:
Bracket Mount

1. Use band saw - 0.125″ Steel plate to roughly 10.5″ × 7″
2. Square off edges of plate using 0.5″ end mill.
3. Mill 3.7″ clearance hole centered at 7.6″ from the short edge of the plate using 0.5″ end mill- keep plate in mill vice.
4. Drill mounting holes for the hydraulic motor as per Appendix V for the shaft side of the hydraulic motor.
5. Remove plate from mill
6. Using new piece of steel cut two 10.5″ × 7.25″ triangles on the band saw.
7. Square off edges of plates using 0.5″ end mill.
8. Cut two 4″ long 90° angle steel using the band saw
9. Drill three 5/16″ clearance holes with the center hole at the center of one of the sides. 1″ spacing for the other two holes.
10. Clean all surfaces using acetone
11. Weld triangles along their long edge to the Steel plate with the majority of hydraulic motor mounting holes oriented at the top of the plate(narrow side of the triangles)
12. Weld angle steel to the inside of each of the 7.25″ bases of the triangles at 1” from narrow edge of the base, holes exposed to the middle of the assembly.

Test Stand:
Standoffs

1. Using chop saw - cut six 7″ long of Bosch Rexroth 45mm aluminum profiles
2. Tap each end on center using 5/16″ UNF tap.

Base Plate

1. Using band saw, roughly cut 0.5″ AL plate to 24″ × 14″
2. Smooth edges with a hand file.
3. Measure 0.875″ from each edge and draw a line across the center of the long edge of the plate
4. Drill a 5/16″ clearance hole at each of these line intersections

Support Hardware:
Belt Tensioner Bracket

1. Use band saw - 0.125″ Steel plate to roughly 4″ × 4″
2. Square off edges of plate using 0.5″ end mill.
3. Drill a mounting hole for the belt tensioner at 2″×3.5″ from a corner

4. Remove plate from mill
5. Using new piece of steel cut two 5″ × 4″ triangles on the band saw.
6. Square off edges of plates using 0.5″ end mill.
7. Cut two 4″ long 90° angle steel using the band saw
8. Drill three 5/16″ clearance holes with the center hole at the center of one of the sides. 1″ spacing for the other two holes.
9. Clean all surfaces using acetone
10. Weld triangles along their short edge to the Steel plate with the majority of belt tensioner mounting holes oriented at the top of the plate(narrow side of the triangles)
11. Weld angle steel to the inside of each of the 5.25″ bases of the triangles at 1″ from narrow edge of the base, holes exposed to the middle of the assembly.

Drive Pulley

1. Use band saw- cut off piece of 5″OD × 4″ long 6061-T6 AL bar stock
2. Fix in lathe chuck and lathe 2″ of the stock to 2″OD
3. Flip the part in the chuck and lathe edge smooth down to 4.6″ OD
4. Use groove tool and lathe down 0.13″ at 1″ from the inside edge of the remaining 4.6″ OD AL.
5. Progress across the exposed face and create 5 more grooves at 0.14″ between groove centers.
6. Using the lathe, drill a 0.375″ hole at the center of the pulley- remove from lathe vice
7. Clamp pulley on mill and find the center of the hole.
8. Zero out the mill coordinates and progress at 90° from center at a distance of 1.565″ from center and drill and tap four 0.75″ deep 5/16″ UNF holes.

Drive Pulley Coupling (Unproven method)

1. Using a CNC mill, 0.375″end mill- on a 4″ square, 0.125″ piece of steel, mill a 4″ diameter circle with a 1.98″OD centered hole-leave part in mill
2. Drill four 5/16″ clearance holes centered on the ring diameter at 90° intervals
3. Remove from mill clamps
4. Press-fit the spline coupling into the ring using a press

Prototype Assembly

1. Referencing Figure 13 above, arrange parts on the base plate.
2. Align all pulleys from the hydraulic motor to the supercharger and mark off the locations for the mounting hole on the base plates.
3. Drill 5/16″ clearance holes in the base plate using the mill
4. De-burr all mounting holes in all the components
5. Attach the leg posts to the base plate and attach the brackets to the base plate
6. Attach the supercharger, hydraulic, and belt tensioner
7. Attach the drive pulley coupling to the splined shaft on the hydraulic motor
8. Attach the smaller pulley (donated) to the supercharger shaft
9. Affix the belt to the supercharger and hydraulic motor pulleys

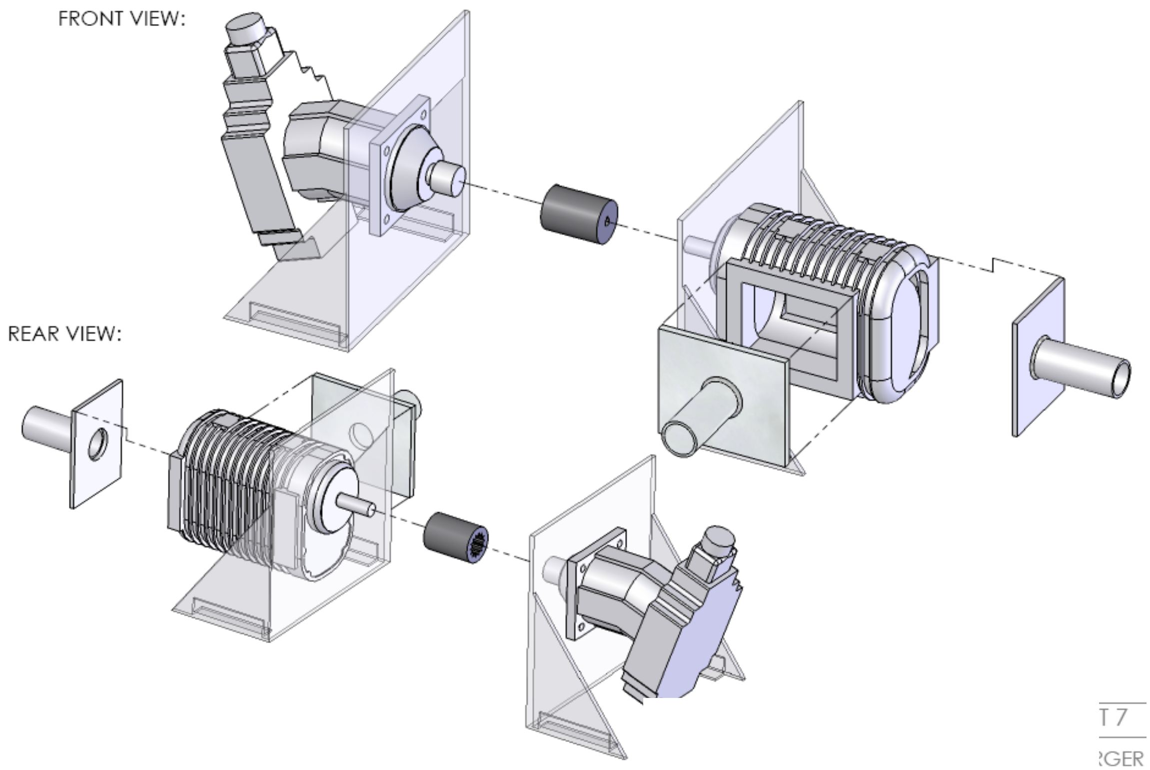
Prototype Issues:
Although we spent a great deal of time fabricating our prototype model, we must stress the need to evaluate the safety of the system. The exposed belt system is a great hazard and does not currently have a guard. Additionally, our spline coupling is manufactured in a way that is not necessarily secure and safe. We do not recommend that this prototype be directly tested as is. A new coupling or a reworking of the current coupling should be performed before the prototype should be tested…or it might explode.

Prototype Costs:
The prototype model costs came to a total of $180.13. The majority of the prototype was allocated from the USEPA with an estimated cost of nearly $7000 just for the hydraulic motor and supercharger. A great deal can be said for the cost of student labor, which is free. However, we as a group spent three weeks nearly 8 hours a day in the machine shop. At 40hr/week for 3 weeks and at an average hourly wage of $35/hour for a qualified technician to do equivalent work, it would cost roughly $4200.

Final Design:
The final design assembly is shown in exploded view in Figure 14. The manufacturing specification differs from the prototype is several ways. The direct coupling of the hydraulic motor to the supercharger removes the need to machine the pulleys for each of the components. This direct coupling also requires modification to the bracket mounts for the supercharger and hydraulic motor. The drive shaft for the hydraulic motor and the supercharger must line up directly to ensure any kind of stability within the assembly.

The spline present on the motor requires that the spline coupling needs to be fabricated with a spline on one side and keyway on the other. This design is outlined in Appendix U with the ECN documentation. Based on previous experience with spline couplings, we can expect this part to cost around $600.

Figure 14: Exploded view of final design



XII. VALIDATION & SIMULATION RESULTS

Physical Observation of Prototype

There are multiple methods that can help to validate that our prototype and final designs work. The first, and most simple, of these methods, is simple physical evaluation of the behavior of the system. To carry out this test, our prototype can be used to evaluate how the system's parts interact. To simulate the power input from the hydraulic motor, we simply grasped the spline coupling attached to the hydraulic motor with our hands, and rotated it quickly. This imparted a rotation to the plate attached to it, which in turn rotated the pulley bolted onto it. The tension in the belt clearly created a zero-slip condition, as shown by the fact that the other pulleys on the supercharger and belt tensioner clearly accelerated at the same time. It was also observable that all three pulleys were very well aligned during this rotation. The next step was to evaluate the output of the system. Putting a hand over the outlet tube of the supercharger, we felt an appreciable air flow and creation of air pressure when we closed off the outlet a little more. All of this was obviously done at very low speeds relative to where the actual system would operate, but it still demonstrated to us that our system worked as desired.

Prototype Validation Testing Plan

The next method of testing is really just the extension of the testing discussed above, except with the proper power and rates applied. Unfortunately we were not able to complete any such testing due to lack of accessibility to the necessary high-pressure reservoir and testing equipment at the USEPA or elsewhere. However, we have developed a method to determine if our designed system meets the desired requirements. This process tests various parts of the system for pressure, flow rate, temperature, and efficiency at the supercharger intake and outlet, and speed at the pulley, with the aid of data acquisition systems currently utilized at the USEPA. If we were able to do such testing, the process we would use would be as follows.

Initial setup of the system will require the direct coupling of several 2″ air hoses rated at greater than 20psi. Additionally no person may be present during operation of the hydraulic motor. The pressures are high enough to possibly cause high pressure pinhole leaks in the conduit hose. No loose clothing, hair, ties, etc. The entire system will be linked in series to the first turbo charger and second turbo charger via 2-inch individual hoses. The outlet from this interchange will connect to the intake manifold of an operating diesel engine.

Use appropriate "-" (dash) hydraulic hose fittings for connecting the hydraulic motor to the high pressure hydraulic accumulator.

Initial Startup and Leak Check

1. The system should be ready once all hydraulic lines are attached to the hydraulic motor.
2. Turning on and Calibrate the data acquisition setup.
3. Obtain baseline values at ambient prior to putting power to the system.
4. Turn on high pressure system and set at 5000 psi with hydraulic motor set at 8cc/rev.
5. Wait for a period of time (1-2 minutes).
6. Shut down high pressure system, slowly bleed pressure and check for leaks.
7. If leaks are present, tighten or replace connectors and repeat steps 4-7 until no leaks are found.
8. Power down system.

Testing for Compliance of:

-Speed up time
-Pressure
-Flow Rate
-Pressure Profile

To test for initial speed up time for max pressure, turn on data acquisition system for pressure, temperature, flow rate, speed, and efficiency.

1. Turn on high pressure supply
2. Begin to record data for a couple seconds before turning on the hydraulic motor, record for 20 seconds maximum or until the max pressure at the supercharger outlet is reached.
3. Analyze the data to find the time to maximum pressure, maximum pressure, and based on the speed, use the manufacturer's specifications to find the flow rate (Appendix Y).
4. Compare results to requested curve for time.
5. Repeat test to verify results.

As listed above, all performance characteristics will be tested for during validation testing.

Prototype Behavior Simulation

The next method of testing, which we did use, really is simply a numerical method of mimicking the testing that we would like to complete as discussed above. There are some limitations to this, but it also provides for a good deal of flexibility in varying parameters that are helpful to have knowledge of prior to actual testing of the prototype, such as the range of displacements to use to maintain steady-state conditions near 5 psi of boost pressure.

To do this numerical simulation, we created a code in Matlab to put together all of the necessary variables and to use a time-stepping method of evaluating the performance of the system, specifically this time-step feature was utilized in being able to mimic the slight delay in the response of the displacement of the hydraulic motor to its input signal. We estimated this delay to be 2 microseconds, and so we used this as our time step throughout the operation of the system, which we simulated for 7 seconds, resulting in a total of 3500 time steps.

At each of these steps, a number of calculations were carried out in order to characterize the system's performance. First, we simulated an input for the pressure difference across the terminals of the hydraulic motor. From this input, we used a control algorithm to calculate the appropriate displacement for the motor based on the pressure difference across the hydraulic motor, the rate of rotation of the supercharger at the previous time step, and the motor's efficiency at the previous time step, as found from the approximated relationship in Figure 17. With this displacement known, this will result in a new set of parameters for the current time step. Specifically, the order of parameters determined will be the torque output from the hydraulic motor and to the supercharger, the power input into the supercharger as found with the rate of rotation from the previous time step, the efficiency of the hydraulic motor and the entire system, the rate of rotational acceleration of the supercharger, the change in rotational speed over the time step, the rotational velocity at the current time step, and the boost output from the supercharger. With all of these parameters determined, the program is then ready to move on to the next time step, and it repeats this process, forming vectors for each variable. The code for this Matlab program can be found in Appendix X. As its output, a number of these variables are plotted against time. All of these plots can be found in Appendix Y, but a few selected important plots are shown below in Figures 15 through 18.

Figure 15: Supercharger Boost and Air Flow versus Time

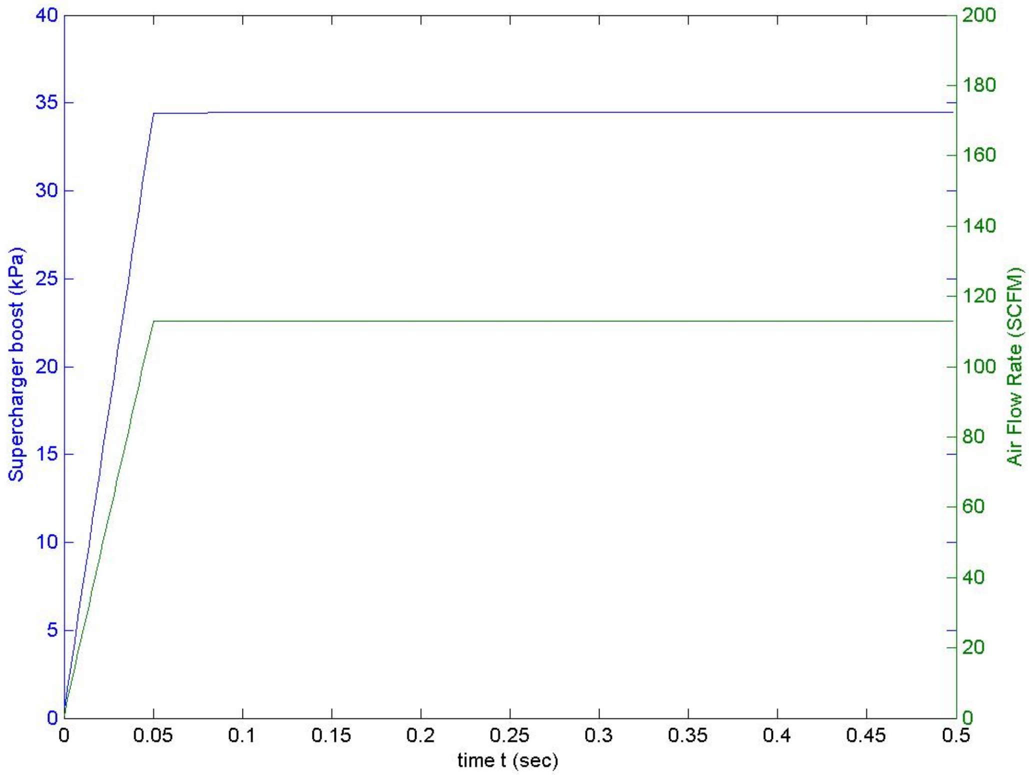


Figure 16: Supercharger Power Input versus Time

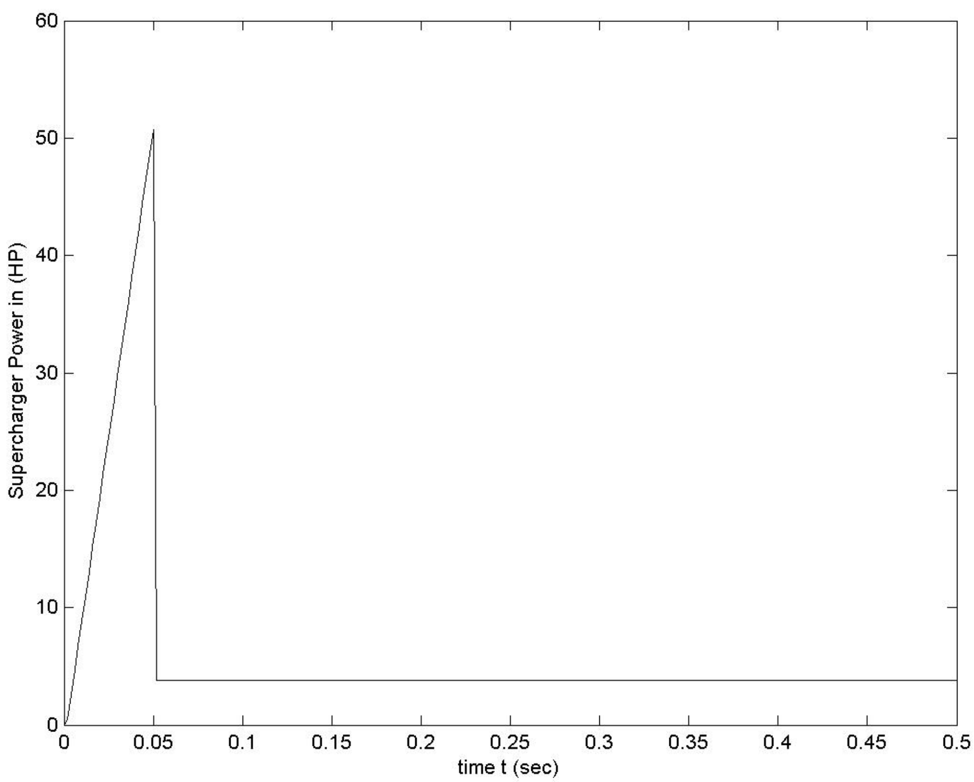


Figure 17: System Efficiency versus Time

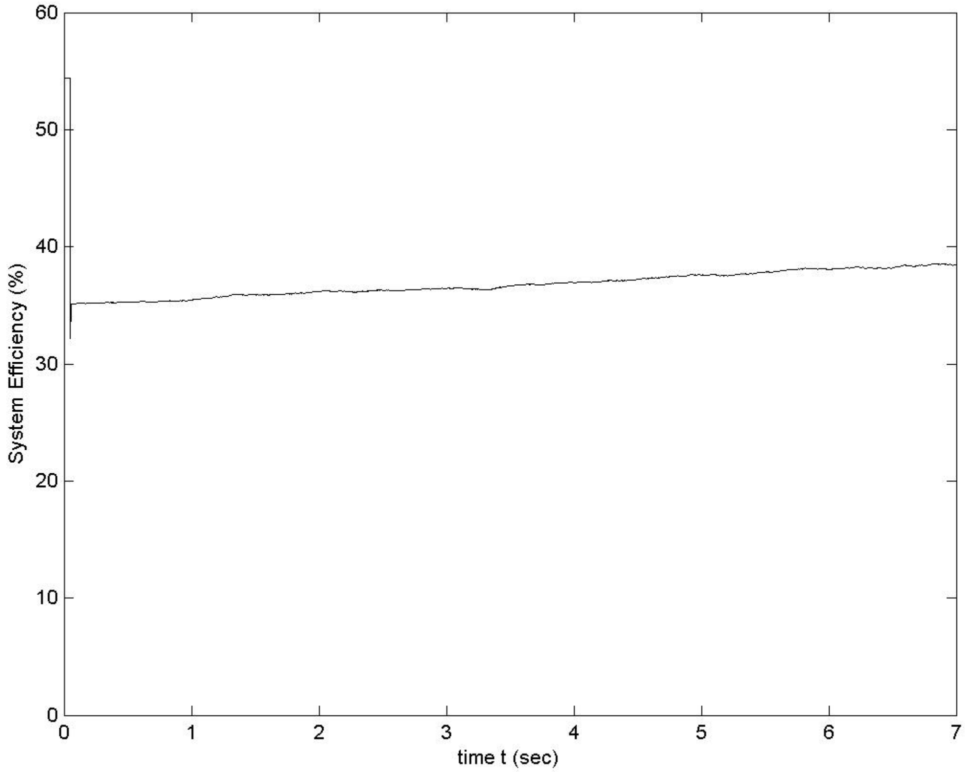
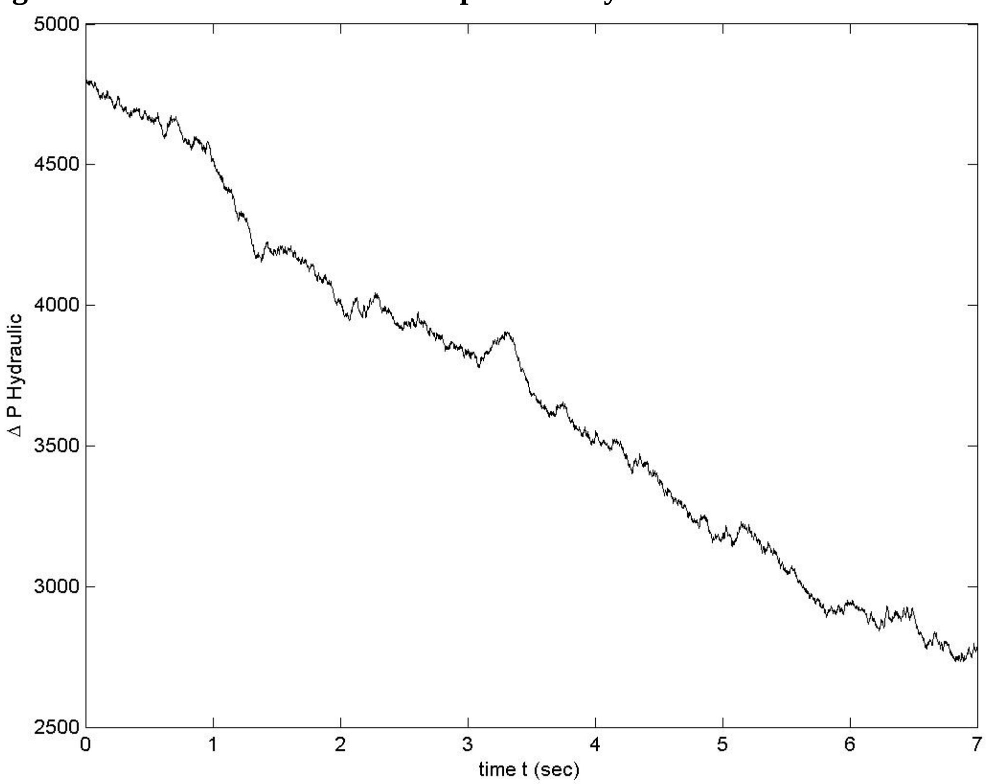


Figure 18: Simulated Pressure Drop across Hydraulic Motor versus Time



We believe that these plots should be a reasonably accurate representation of the performance of the system due to the fact that the data has been derived from published equations and known performance specifications. Also, we did take into account the necessary real factors that influence the system, such as the rotational inertia of each rotating component in the system, most notably the pulleys. Of course, all of our numerical results still represent estimations since there will always be experimental differences such as friction that causes additional efficiency losses, but we believe that this behavior will be shown by the prototype system. One key difference between these results and those of our final design is that with a smaller, custom-designed hydraulic motor, the speed-up time will be slower. To stay within the limit of 0.27 seconds for a speed-up time, the maximum displacement of the hydraulic motor should not be any smaller than approximately 7 cubic centimeters, which is one quarter of the maximum displacement in our prototype's hydraulic motor. As long as this limit is observed, the hydraulic motor should be designed to be as small as possible in order to maximize efficiency. This numerical simulation, in conjunction with testing of the actual prototype, should fully validate the viability of our final design.

XIII. ENGINEERING CHANGES NOTICE (ECN)

There were several changes between the final design, as documented in Design Review #3, and the actual prototyped part. A detailed listing of Engineering Change Notices (ECN's) can be seen in Appendix U. For example, a platform of height seven inches was added to the mounting plate of the supercharger system that will allow the mounting plate to clear plumbing that is present on the USEPA test cart. In addition, the team decided to use a variable displacement hydraulic motor instead of a fixed displacement motor, which will allow for accurate control of system output.

In terms of the mounting pieces for the supercharger system, it was decided to have the hydraulic motor, belt tensioner, and supercharger mount to individual brackets rather than a single bracket. This will allow for flexibility in aligning the components. Another ECN that led to flexibility in aligning components was the use of L-brackets for mounting pieces rather than welding them to the mounting plate.

To make the supercharger system compact in size, the belt tensioner was moved from above the belt to below it, which reduces the overall height of the system. To reduce the manufacturing time on the prototype, the outlet manifold was simplified such that the aluminum tube is welded to the manifold plate rather than in a tapered form. Finally, the hydraulic motor pulley was re-designed so that, rather than directly connecting the spline connector, it can be connected via a plate interface.

XIV. DISCUSSION

Discussion of Project Planning and Accomplishments

Now that we are done with our project, we can look at the process we used to obtain our final product, and see how we could have and should have done some things differently. One major difference that we see is that we should have immediately gone to the USEPA about the availability of components, rather than doing so much research into outside components such as the Sea-Doo supercharger or the possibility of using another type of compressor such as an air-conditioning compressor from a junk yard (we took a trip to Dundee, Michigan). This extra research time spent was wasted in the end, but at least was informative in terms of what possibilities might be useful in a similar project that might be of a different scale.

Another thing that we should have done is to complete prototype validation testing at the USEPA or elsewhere, provided that we had sufficient time to do so. The main source of time that we could have used to do this testing would have needed to come from the wasted time early in the project looking into other options such as the Sea-Doo supercharger. This lack of testing data could be considered as the largest weakness in our project as a whole, since it would provide a lot of data which would contribute in validating the theory. However, we do have a replacement for this testing in our numerical simulation in Matlab, which should be a reasonable approximation of the prototype system's actual performance.

Discussion of Design

Our designs for both our prototype and our final design are well thought out and fully engineered to the best of our team's abilities. We completed all of the necessary research to understand the operation of our system, and to understand all of our possible design variations. We selected these ideas carefully, with every decision being made with sound reasoning, not arbitrary decision-making. For example, in order for our prototype to meet the needs of the USEPA, we needed to raise the base plate seven inches above the ground level on six legs in order to accommodate the appropriate plumbing needed to run the system in the laboratory environment. We consider the carefully-chosen nature of the system's parameters and dimensions to be one of the major strengths in our project.

Despite our final prototype design being along the lines of what we want, we still should have made a number of changes to the way we went about our design process. We chose a belt that seemed like a reasonable length and then designed the separation between our components around this. This should have been done opposite to this, with the separation estimated first, and then a belt length chosen from this. Another change is that we should have put more thought into the thickness of the supercharger manifolds, as we used half-inch-thick aluminum plates, while the USEPA has manifolds that are significantly thicker and made of steel. We should have looked into the reasoning behind this, and also should have looked at whether these blocks were hollowed out with a tapered surface used to funnel the air flow towards the inlet and outlet tubes more efficiently.

This entire design process led us to a prototype solution that is quite satisfactory to us because it provides not only for the opportunity to test a full-scale model of the final design, but also allows for precise manipulation of the displacement of the hydraulic motor in order to fully characterize system performance. Thus one of the major strengths of our prototype is its flexibility in testing parameters.

There are many aspects of the prototype design that can be improved. Based on the required pressure/flow profile provided by the USEPA, performance parameters were determined. Next, research was conducted in determining which hydraulic motor and supercharger most efficiently met the performance requirements. However, the USEPA provided us with a hydraulic motor and supercharger free-of-charge, and these were used in the prototype design, despite the fact that we would need to run them significantly below their ideal operating points. For future modifications of the prototype design, improvement can be made with respect to choosing a hydraulic motor and supercharger that can be running at or near its ideal operating point while achieving the desired pressure/flow profile.

Discussion of Manufacturing

We should have changed a few machining processes to some degree. One such change is that for the large pulley, we should have obtained a piece of smaller aluminum stock such that the initial lathe operation would have taken significantly less time. Another thing we could have done differently is that we should have used the CNC mill for all of the large holes that we made, rather than using a manual rotating table. This not only would have saved us a lot of machining time, but would have saved us a lot of effort, with the additional benefit being an improved surface finish on the outside edge. In addition, it would have given us more time on the manual mill either for ourselves to be doing additional machining that we needed to do at that time, or for other groups to use.

In terms of the dimensions of our prototype, what might appear to be the most arbitrary decisions were still based on our best engineering judgment. For example, the distances between mounting holes in the L-brackets that attach the support structures to the base plate were selected for convenience, but were still selected with our judgment that they should be symmetric about the middle point, and should have sufficient bolt head clearance of 0.5 inches off of the edges of the brackets.

Once we had a good concept of what we were going to be building, we did not always have engineering drawing of the parts we would be making, but we had a good understanding of their functions, so we did not end up wasting much material in re-making parts that did not fit our needs. The largest such part was a circular mounting disc for the hydraulic motor's pulley, which was made too small in the outside diameter because we did not account for the mounting system moving, resulting in a disc that was continually getting smaller as we milled the outside edge.

A number of changes to the manufactured prototype would have been helpful, if we had more time to work on it. One thing that we should have done differently is to use a different way of attaching the spline coupling on the hydraulic motor to the pulley, since we were unable to weld together two pieces of dissimilar steel materials. This is obviously then a literal weakness in the design.

Another thing we should have done differently is the entire setup of the belt tensioner. It was not properly designed, and then was welded such that it did not account for this error. Specifically, it needed to be higher off of the base plate, so the initial triangle supports cut out should have been taller. As a result, we had to add an additional half-inch slab of aluminum on the base plate to raise it sufficiently. Also, the tensioner was not positioned correctly on the base plate when we drilled the mounting holes, so slots had to be milled to make room for adjustability in this regard.

XV. RECOMMENDATIONS

In terms of recommendations for our sponsor, Andy Moskalik, we need to restate a few verbal recommendations that we gave at the design expo. Specifically, the spline coupling and the plate it is attached to need to be modified to have a stronger connection since right now they are only held together by a press-fit. All appropriate safety devices need to be added for protection of the pulley and belt system since this could be dangerous during operation. These components are detailed in section XI on the manufacturing plan.

The most challenging engineering aspect that we had with our project was meeting the pressure and flow requirements for our boost curve. Utilizing traditional boost mechanisms, i.e. a supercharger, we were able to produce this low pressure at a proper flow rate. However, this was achieved at a great loss in efficiency for both the hydraulic motor and the supercharger. It might be worthwhile to engage the pressure profile a little more thoroughly and develop a better understanding of not just the experimental setup of the test cell, but also the data acquisition process. This could possibly lead to a design compromise in the basic requirements for the project resulting in a more efficient system prototype from the next student group.

To future ME 450 design teams, we would recommend further development of our ideas towards the goal of our final design. Specifically, it would be beneficial not only to find a way to optimize the size of the hydraulic motor, but to actually design it, at least as far as displacement. It may also be useful to develop a system for this process such that the optimal hydraulic motor may be easily determined from the parameters of the supercharger attached to it. Additionally, our final design recommendation would need research into what would be required in order to have an exact 1:1 drive ratio between the hydraulic motor and the supercharger. In doing such research, the teams that work on this project should realize that they should not hesitate to ask their sponsors for help and advice.

XVI. CONCLUSIONS

The goal of this project is to design, verify, and build an on-demand supercharger, powered by a small hydraulic motor, to produce additional boost for mid-range speeds according to the desired pressure profile for 6 to 7 seconds. Our design should be durable, efficient, and quick to power up. The reason for the use of a hydraulic motor is that in order to minimize power deduction from the engine, efforts are being made to utilize pressurized transmission fluid. Current fluid-powered superchargers, known as hydrachargers, do not meet the customer requirements for both efficiency and performance control. The significance of this project is that if we are able to accomplish our goals, we will have created an efficient on-demand supercharger that could readily be adapted to use with a hydraulic hybrid chassis. Since this type of supercharger does not currently exist, the success of the project could lead to patent application and production use.

The on-demand hydraulic supercharger system has presented a fair amount of technical challenges. The team has developed engineering specifications based on the customer requirements provided by the USEPA. Performance equations for the supercharger, belt drive, and hydraulic motor were determined, and power requirements were found for each component, accounting for efficiencies. A prototype design was created with a hydraulic motor and supercharger on-loan from the USEPA, and the team manufactured mounts for the system as well as the pulleys for the belt system. A final design can be determined based on the successful validation testing of the prototype. The team has created a validation plan for testing the prototype at the USEPA. In addition, the team has created a Matlab simulation of system performance which approximately meets the pressure/flow profile provided by the USEPA.

The approximate efficiency of the prototype on-demand supercharger system is low, 33.6%. This is due to the combined lower efficiencies for both the motor and the supercharger. Low efficiency has been expected because of the over-designed nature of our prototype. Both the motor and the supercharger are intentionally under-powered to meet our design requirements. It is important to reiterate that the driving factor for supercharger selection was the ability of the screw type supercharger to produce boost immediately regardless of input speed. However, the result of using such a method is that both components operate at less than optimal efficiencies.

Our group recognizes that the system can be improved. The prototype we have developed will utilize a supercharger and hydraulic motor that are already in existence. Both of these components can theoretically produce the required pressure profile. However, after extensive research we have come to the conclusion that there is no existing ideal solution to the problem at hand. We feel there is no known ideal variable-displacement hydraulic motor to meet our high fluid pressure requirement. Given this condition, we are confident that our system will perform as required, but will need to be reevaluated for efficient use for higher pressure and flow applications. The final design offers one possible solution to further raise system efficiency and reduce the number of components within the system.

When considering future applications of the prototype system, it will be important to characterize the prototype system's operating point of maximum efficiency. This will establish final feasibility for directly coupling the supercharger with the hydraulic motor. However, as mentioned previously, it is imperative that the coupling to the drive pulley be revised so that the system can be run safely at operating speeds.

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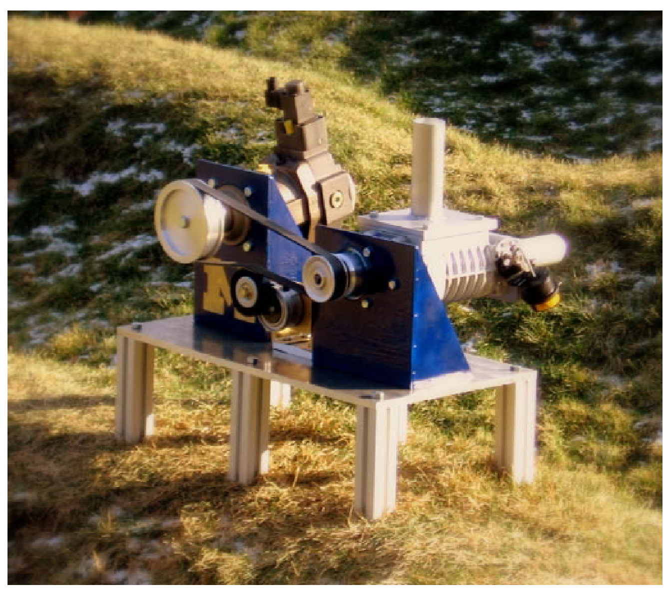

XVIII. INFORMATION SOURCES

In order to gain background knowledge on the operation of superchargers and hydraulic motors, the team gathered information regarding our project from discussions with our sponsor and consultation with a small start-up company, as well as online sources [3, 5, 10, 11].

In terms of engineering resources, the USEPA provided us with a pressure and flow rate versus time profile that they have requested us to meet, which may be found in Appendix D. The profile is based on an existing hydracharger that was developed by the USEPA. However, the existing prototype has several problems with regards to efficiency and control, which is why the need for a new prototype exists. From the profile, we were then able to begin research the types of blowers and superchargers that could provide the necessary pressure with reasonable RPM and torque requirements based on the available hydraulic motors.

Our initial top candidate was a supercharger designed for Sea-Doo watercrafts by RIVA Motorsports (RIVA ProSeries SD Supercharger). However, we finished with an Eaton MP45 Supercharger loaned to us by the USEPA, free of charge. One helpful aspect of this supercharger was that it has published performance data online which we were able to use in determining the design of our system. Specifically, this data allowed us to find the necessary torque and RPM levels to efficiently power the supercharger.

Obtaining the specifications of the custom hydraulic motor for the final design is the largest information gap we currently have. This information will come largely from analysis of the results gained from the eventual testing of our prototype system at the USEPA. Once this is done, contact with manufacturers may be able to lead to the development of a custom motor or selection of a motor that we do not yet know exists.

We have collected a number of technical resources dealing with the determination of compressor and motor requirements. We used *Compressor Performance*, by M. Theodore Gresh, to determine the type of supercharger that best suits our needs [10]. Since our customer requirements include both durability and efficiency, we must consider the tradeoffs between the two. Specifically, Table 4 in the concept generation and selection section indicates that the efficiency of the axial compressor is higher than that of the centrifugal compressor, but only runs at higher speeds. Thus the axial compressor might appear to be preferable, but other factors must be considered. The fact that the axial compressor runs at higher speeds means that it will produce a lower torque, given the same power input. Also, the axial compressor is known to have a fragile and expensive set of blading, thus the customer requirements for durability and low cost would not be met.

This leads us to consider the use of a centrifugal compressor for our supercharger. It is known that the pressure of a centrifugal compressor is roughly constant with variable flow, which should make the supercharger easier to control. It does have a higher efficiency than a screw-type positive displacement compressor, as well as lower weight and increased capacity. However, we may use the screw-type compressor due to cost, since the USEPA already has this type of supercharger. One other option would be an ejector compressor, but this type has low efficiency and requires a high-pressure source. The centrifugal compressor does not require such a source, as it functions with an ambient air pressure source.

As for marketing resources, the USEPA introduced us to a small start-up company called Blizzard Boost, who manufactures on-demand engine performance via super-chilled, high-density oxygen injection into the combustion chamber. Our team met with the President and Vice President of Blizzard Boost and discussed the high demand for low-cost, high-efficiency increases in engine performance. Blizzard Boost specifically is interested in appealing to teenagers and young adults looking to modify their cars, whereas the USEPA is focused on the market of delivery truck companies looking to improve the fuel economy of their fleet.

Our team did further research online into the market for on-demand hydraulic superchargers, and came across the Garrett® HydraCharger™, an on-demand hydraulic turbocharger manufactured by AlliedSignal Turbocharging Systems [2,4]. The Garrett® HydraCharger™, as seen in the drawing in Appendix E, was marketed to passenger car and truck manufacturers who are looking to reduce emissions and improve fuel economy to meet environmental regulations. The Garrett® HydraCharger™ was first introduced in 1998, but due to its gravity drain requirements and inefficiency in fluid energy transfer to the supercharger, it was discontinued [6,7].

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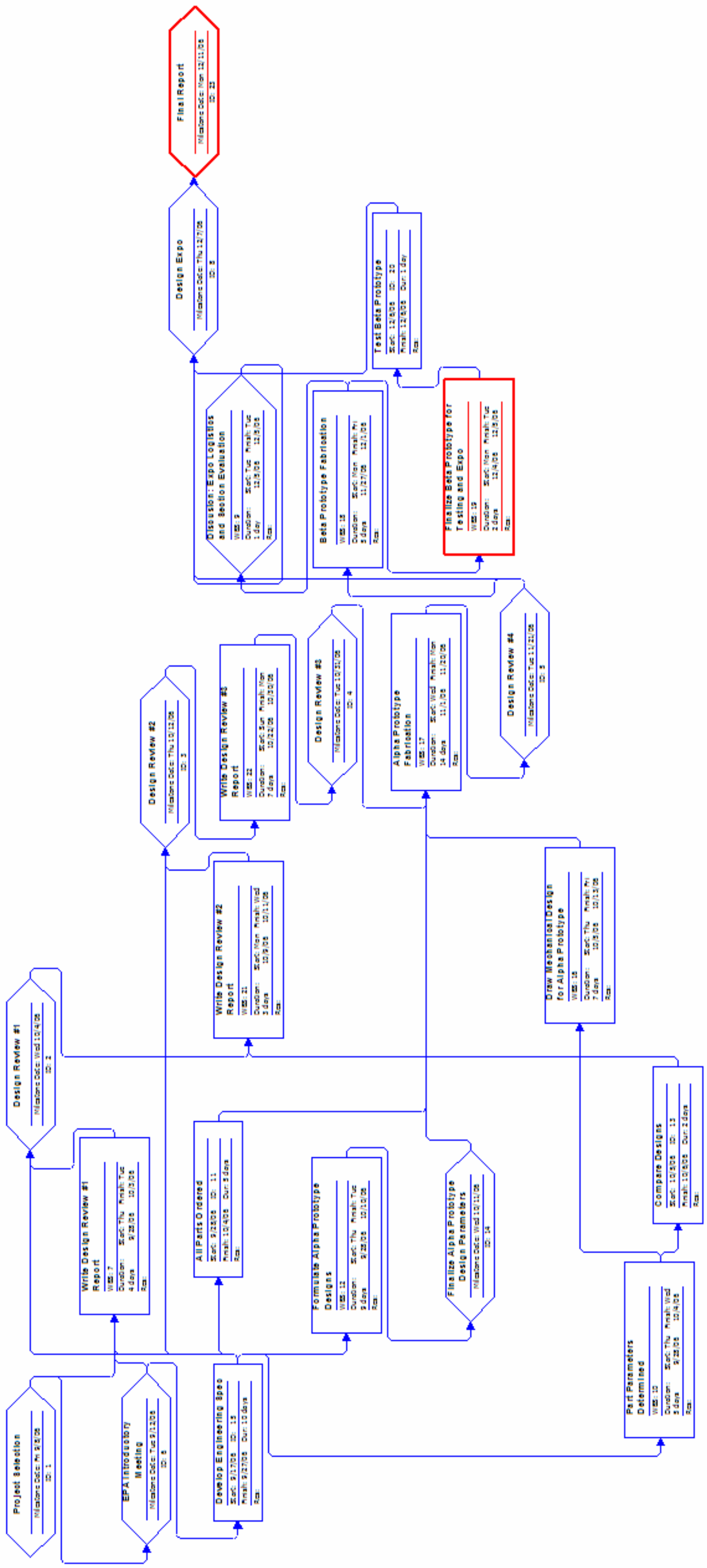
XX. APPENDICES

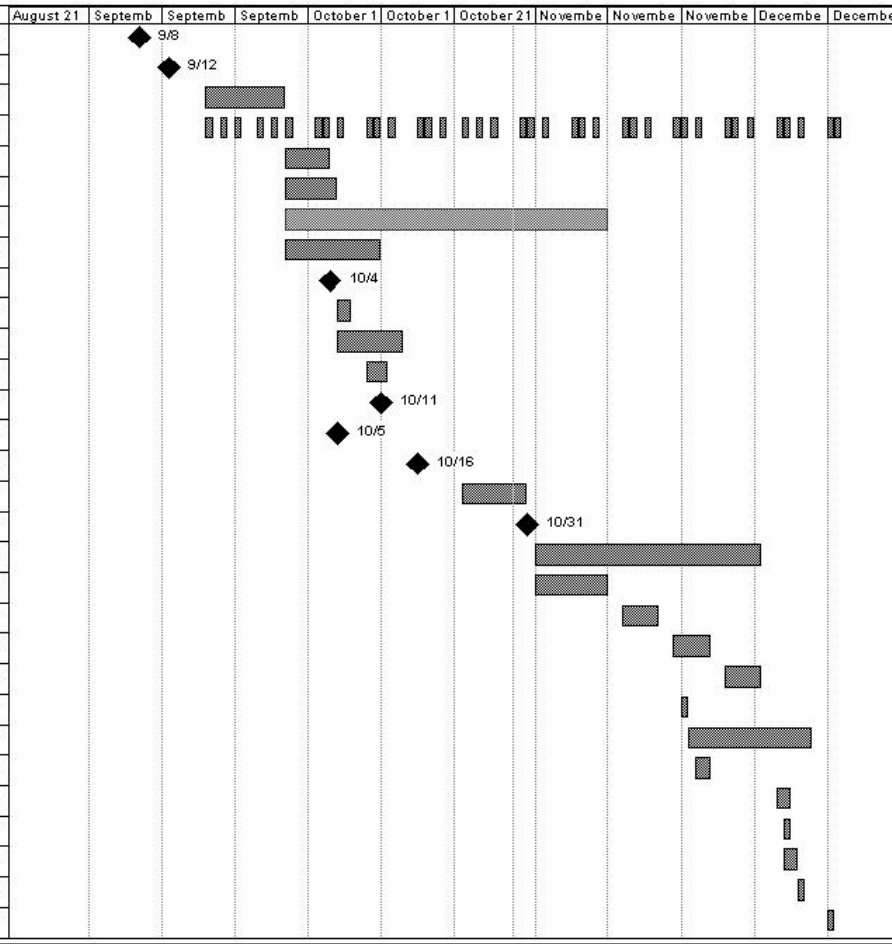
Appendix A: Pert Chart



Appendix B: Gantt Chart

Gantt Chart Page

ID		Task Name	Duration	Start
1		Project Selection	1 day	Fri 9/8/06
5		EPA Introductory Meeting	1 day	Tue 9/12/06
12		Develop Engineering Specs for Build	10 days	Sun 9/17/06
16		Weekly Group Meeting	65 days	Sun 9/17/06
6		Write Design Review #1 Report	4 days	Thu 9/28/06
7		Part Parameters Determined	5 days	Thu 9/28/06
8		All Parts Ordered	33 days	Thu 9/28/06
9		Formulate Alpha Prototype Designs	9 days	Thu 9/28/06
2		Design Review #1	1 day	Wed 10/4/06
10		Compare Designs	2 days	Thu 10/5/06
13		Draw Mechanical Design for Alpha Pr	7 days	Thu 10/5/06
14		Write Design Review #2 Report	3 days	Mon 10/9/06
11		Finalize Alpha Prototype Design Pa	1 day	Wed 10/11/06
3		Design Review #2	1 day	Thu 10/5/06
55		Fall Break	2 days	Mon 10/16/06
15		Write Design Review #3 Report	7 days	Sun 10/22/06
4		Design Review #3	1 day	Tue 10/31/06
56		Beta Prototype Fabrication	23 days	Wed 11/1/06
57		Manufacture Belt/Pulley Drive	8 days	Wed 11/1/06
58		Manufacture Tensioner for Belt	5 days	Mon 11/13/06
59		Manufacture Compressor Manifold	5 days	Mon 11/20/06
60		Manufacture Boost Control Mechanis	5 days	Mon 11/27/06
61		Design Review #4	1 day	Tue 11/21/06
62		Prepare Final Report	13 days	Wed 11/22/06
63		Thanksgiving	2 days	Thu 11/23/06
64		Assemble and Finalize Beta Prototyp	2 days	Mon 12/4/06
65		Discusion: Expo Logistics and Sectio	1 day	Tue 12/5/06
66		Test Beta Prototype	2 days	Tue 12/5/06
67		Design Expo	1 day	Thu 12/7/06
68		Final Report Due	1 day	Mon 12/11/06

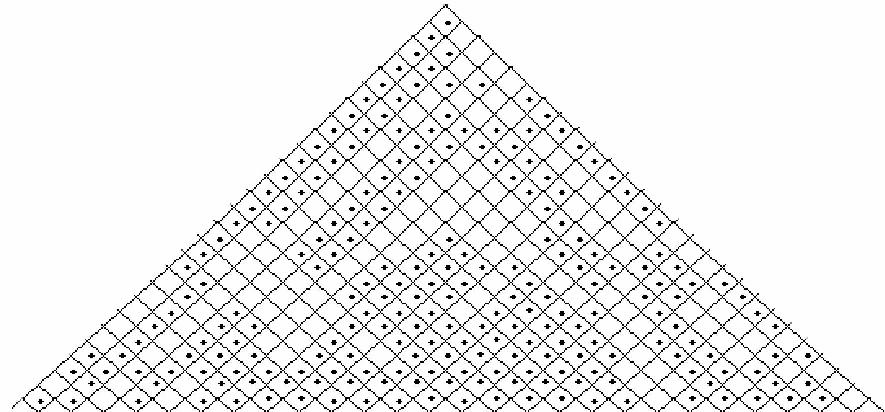


Project: Project7
Date: Sun 10/29/06

Task · Split · Progress · Milestone · Summary · Project Summary · External Tasks · External Milestone · Deadline

Page 1

Appendix C: QFD Chart

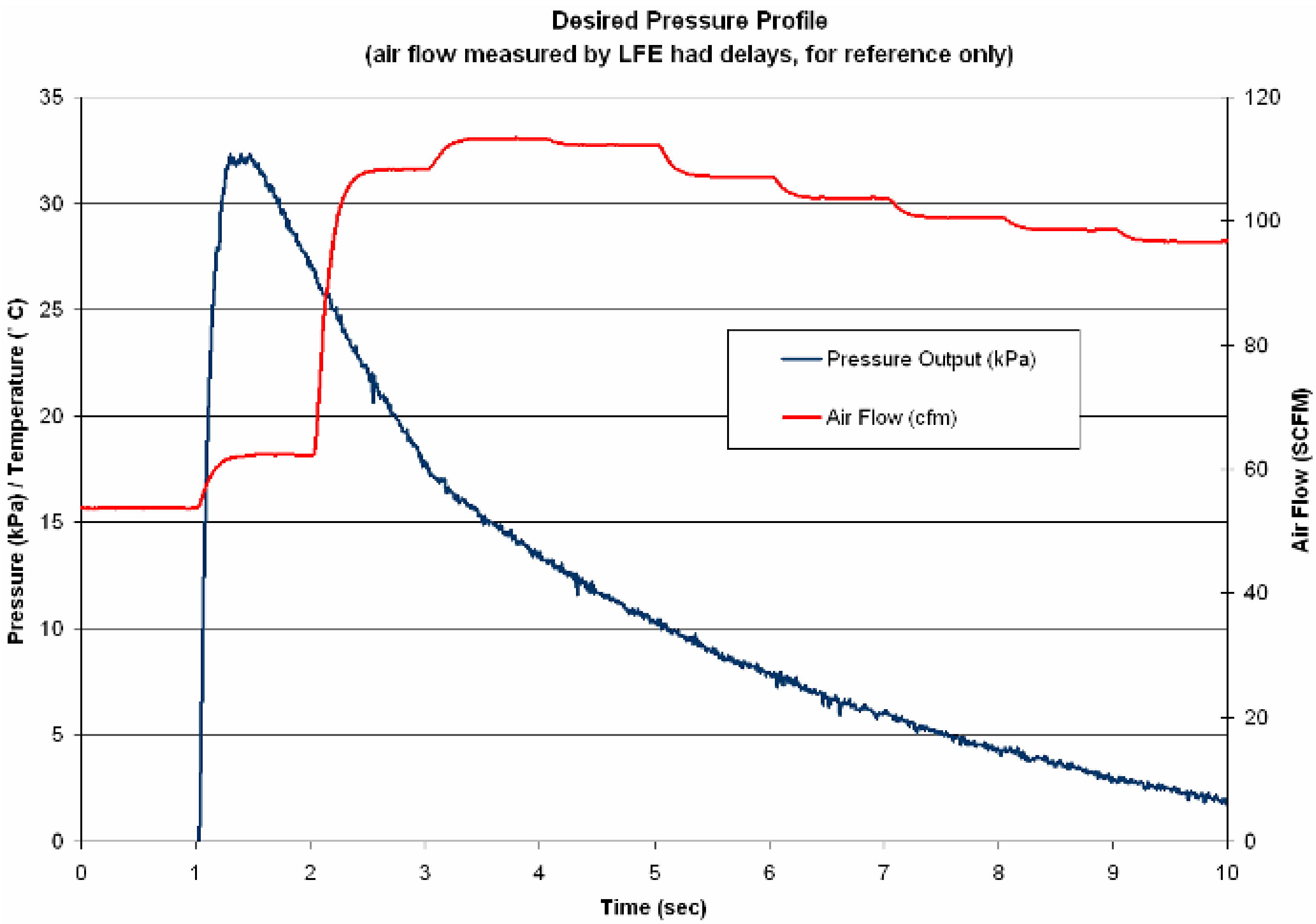


1	**Not related**
3	**Weakly related**
5	**Neutral**
7	**Moderately related**
9	**Strongly Related**

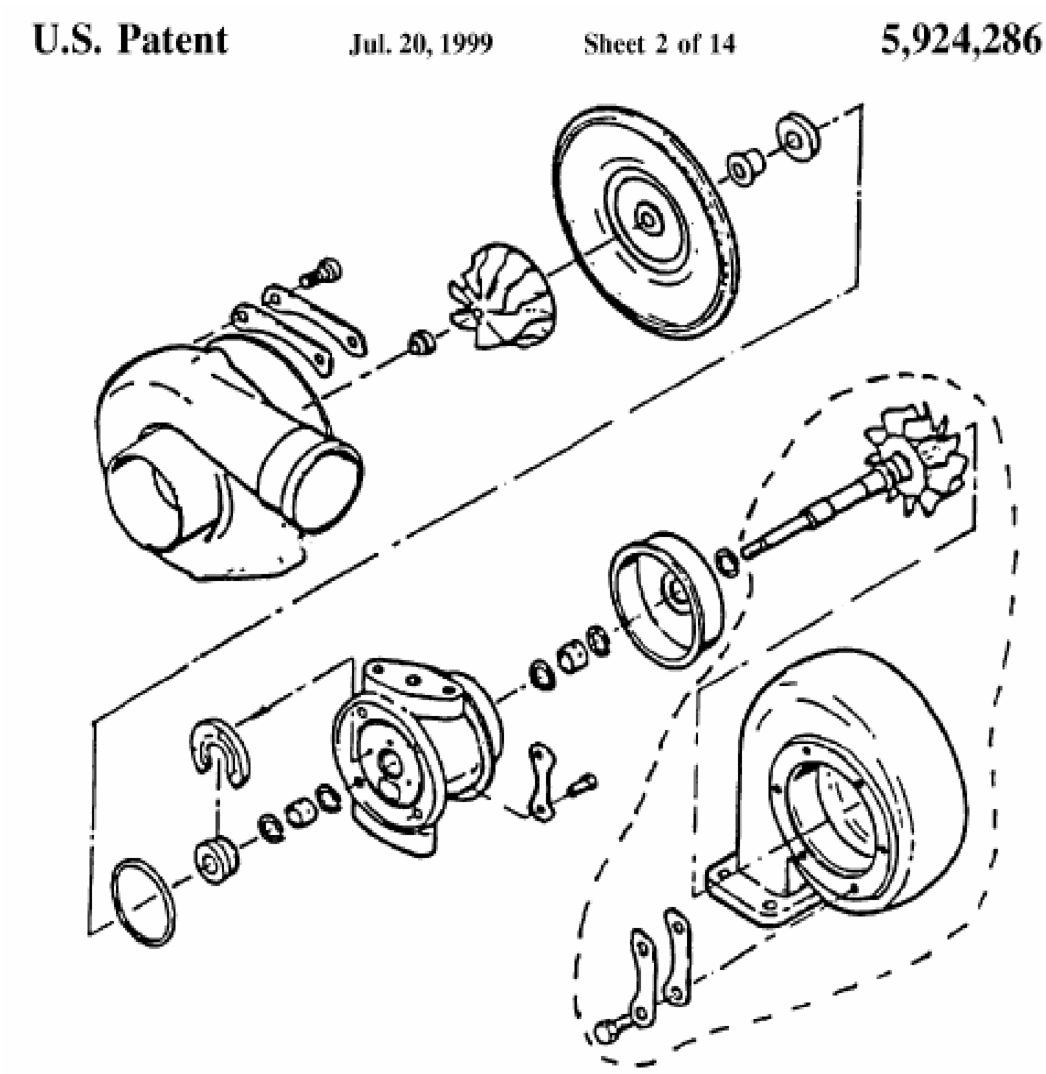
USER-PERCEIVED QUALITY

CUSTOMER REQUIREMENTS (USEPA) / ENGINEERING SPECS	Normalized Importance to Customer (Relative Weight)	Manufacturing & Development cost	Max Boost Pressure at Supercharger Outlet	Ambient Air Pressure at Supercharger Inlet	Supercharger Flowrate	Supercharger RPM	Supercharger Intake Diameter	Supercharger Outlet Diameter	Supercharger Size Length	Supercharger Size Width	Supercharger Size Height	Coupling Unit RPM/Torque conversion	Coupling Unit Length	Coupling Unit Width	Coupling Unit Height	Hydraulic Motor Operating pressure	Hydraulic Motor RPM Output	Hydraulic Motor Torque Output	Hydraulic Motor Size Length	Hydraulic Motor Size Width	Hydraulic Motor Size Height	Material strength, stiffness, durability	No leakage of air/hydraulic fluid	Engine to Supercharger, Hydraulic Motor to High Pressure Reservoir Interface	System Mechanical Efficiency (friction)	System Thermal Efficiency (heat generation)	System Pressure Output Control	System Weight	Total - Customer Requirements	Rank	Importance	Benchmarks	Garrett Hydracharger
Durable	0.75	3	1	1	1	1	1	1	1	1	1	7	5	5	5	1	1	1	1	1	1	9	9	9	9	7	1	3	65	8	0.048		9
Efficient	0.90	7	7	1	9	9	3	3	3	3	3	9	3	3	3	9	9	9	3	3	3	7	9	7	9	9	7	7	141	3	0.104		3
Quick to power up	0.95	3	5	5	7	9	5	5	3	3	3	9	3	3	3	7	7	9	5	5	5	1	5	7	9	7	5	3	134	4	0.099		9
Meets pressure profile provided	1.00	7	9	7	7	9	5	5	3	3	3	9	3	3	3	9	7	9	5	5	5	3	7	7	7	7	9	3	159	1	0.117		9
Meets flow profile provided	1.00	7	7	7	9	9	5	5	3	3	3	9	3	3	3	7	9	7	5	5	5	1	5	7	7	7	7	3	151	2	0.111		9
Inexpensive	0.55	9	1	1	7	7	1	1	1	1	1	1	1	1	1	5	7	7	7	7	7	7	5	5	7	5	3	7	62	10	0.046		3
Recycles hydraulic fluid	1.00	9	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	7	7	7	3	9	9	3	1	7	1	79	8	0.058		5
Appearance	0.20	7	3	3	3	3	9	9	9	9	9	9	9	9	9	3	3	3	9	9	9	5	5	9	3	3	7	9	35	15	0.026		3
Uses existing parts	0.80	9	1	1	1	1	1	1	3	3	3	3	1	1	1	5	3	3	3	3	3	3	5	9	9	7	7	5	76	9	0.056		1
Works in pre-existing hydraulic system	1.00	9	1	1	1	1	5	5	3	3	3	1	1	1	1	7	5	5	3	3	3	3	3	9	7	7	5	1	97	6	0.071		9
Easy to manufacture	0.50	9	1	3	1	1	1	1	3	3	3	7	3	3	3	1	1	1	1	1	1	5	3	9	7	5	3	5	43	12	0.031		5
Easy to repair	0.45	7	1	1	1	1	1	1	3	3	3	7	3	3	3	1	1	1	1	1	1	5	5	9	7	5	3	1	36	14	0.026		3
Simple user interface	0.55	9	9	1	1	1	1	1	1	1	1	9	1	1	1	1	1	1	1	1	1	1	1	7	1	1	9	3	37	13	0.027		5
Low Weight	0.30	5	1	1	1	1	7	7	7	7	7	7	7	7	7	1	1	1	7	7	7	9	7	9	7	7	3	9	44	11	0.033		5
Easy to install	0.65	7	3	3	3	3	7	7	5	5	5	7	5	5	5	1	1	1	5	5	5	5	9	9	7	7	7	7	90	7	0.067		5
Compact Size	0.65	7	3	3	7	7	7	7	7	7	7	7	7	7	7	5	5	5	7	7	7	7	7	9	3	3	3	7	107	5	0.079		3
Units		$	kPa	kPa	cfm	rpm	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	-	[illegible]	[illegible]	[illegible]	kPa	rpm	N-m	[illegible]	[illegible]	[illegible]	[illegible]	cfm	-	[illegible]	[illegible]	kPa	[illegible]					
Now		0	TBD	TBD	TBD	TBD	TBD	TBD	TBD	TBD	TBD	TBD	TBD	TBD	TBD	TBD	TBD	TBD	TBD	TBD	TBD	TBD	TBD	TBD	TBD	TBD	TBD	TBD					
Garrett Hydracharger		TBD	[illegible]	[illegible]	[illegible]	TBD	TBD	TBD	TBD	TBD	TBD	TBD	TBD	TBD	TBD	TBD	TBD	TBD	TBD	TBD	TBD	TBD	TBD	TBD	TBD	TBD	TBD	TBD					
Target		[illegible]	[illegible]	[illegible]	[illegible]	TBD	TBD	TBD	TBD	TBD	TBD	TBD	TBD	TBD	TBD	TBD	TBD	TBD	TBD	TBD	TBD	TBD	TBD	TBD	TBD	TBD	TBD	TBD					
Total		[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]					
Rating (%)		[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]					
Ranked Importance		2	18	27	15	11	19	19	21	21	21	4	24	24	24	8	10	9	12	12	12	15	5	1	3	6	7	17					

Appendix D: Pressure/Flow versus Time Profile



Appendix E: Garrett® HydraCharger™

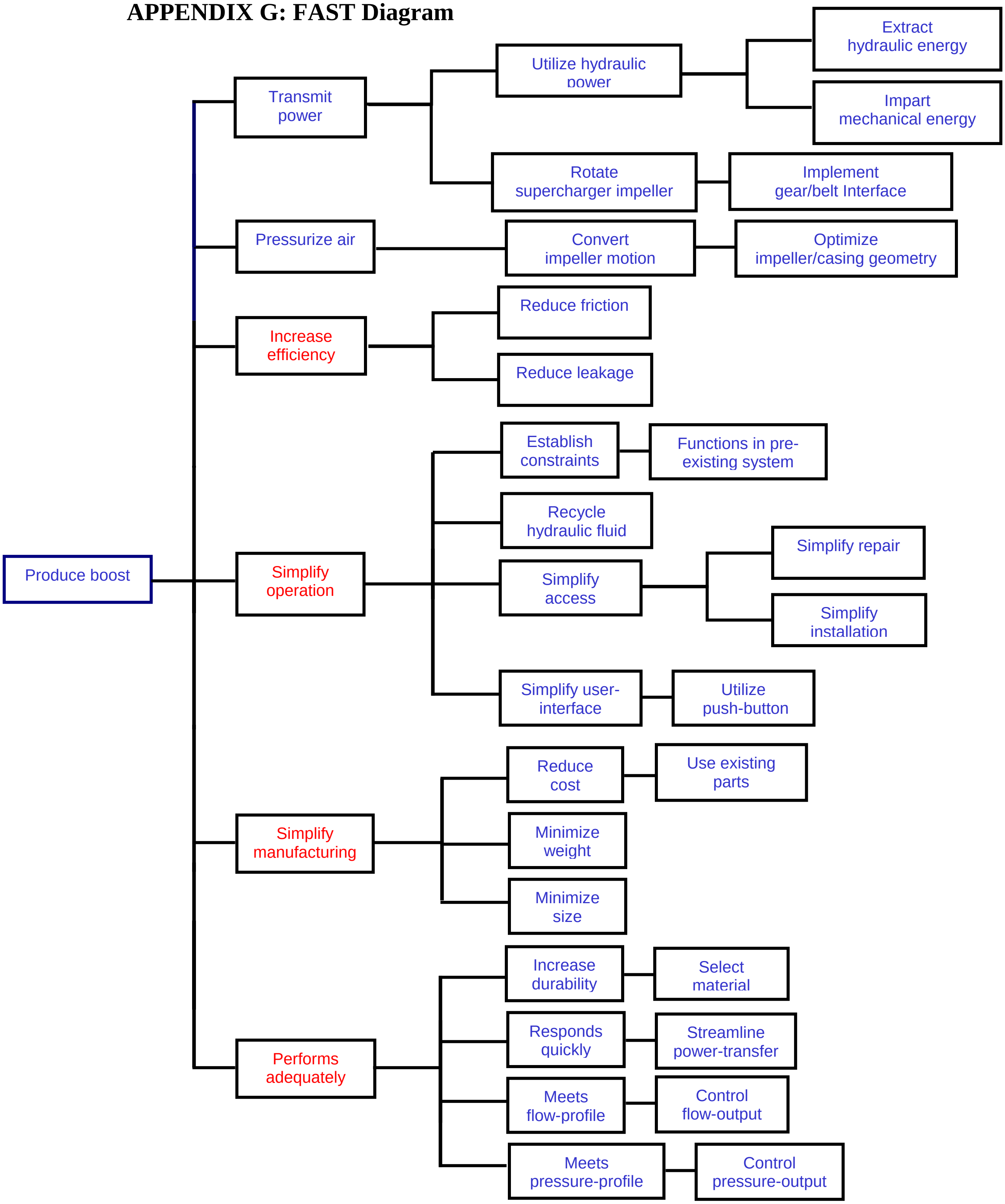


Appendix F: RASIC Chart

RASIC Chart: 11/13/06-12/11/06

Title	Responsible	Start Date	End Date	Time Required (%)	Support	Issue Description	Concur
Acquire Prototype Materials	All	11/13/2006	11/14/2006	5%	Alro Metals Plus (734) 213-2727 2282 S. Industrial Highway Ann Arbor MI 48104	Buy aluminum plates and pipes for supercharger manifolds as well as system mount	Complete
Manufacturing Plan	All	11/15/2006	11/16/2006	5%	Bob Coury, Marv Cressey	Develop manufacturing plan that includes dimensional drawings for all components, as well as feed rates and cutting speeds for mill and lathe operations	Complete
Prototype Manufacturing	All	11/17/2006	12/5/2006	50%	Bob Coury, Marv Cressey	Manufacture the supercharger manifolds, mounting pieces for the hydraulic motor, belt tensioner, and supercharger, system mounting plate, hydraulic motor pulley	Complete
Prototype Assembly	All	12/6/2006	12/7/2006	5%	Bob Coury, Marv Cressey	Obtain appropriate fasteners, clean all surfaces prior to welding, weld supercharger manifolds and brackets onto the mounting pieces. Sand surfaces and paint.	Complete
Prototype Validation Plan	All	11/20/2006	12/7/2006	5%	US EPA: Andy Moskalik (734-214-4719)	Develop prototype validation plan prior to testing to establish proof-of-concept for the final design	Complete
Prototype Validation Testing	All	12/7/2006	TBD	-	US EPA: Andy Moskalik (734-214-4719)	Attach supercharger system to test cart at the US EPA, and attach pressure, temperature, and flow reading devices to appropriate locations	Incomplete
Design Review #4	All	11/14/2006	11/21/2006	10%	US EPA: Andy Moskalik (734-214-4719)	Present prototype-in-progress to Prof. Saitou and Prof. Skerlos	Complete
Design Expo	All	12/1/2006	12/7/2006	10%	US EPA: Andy Moskalik (734-214-4719)	Make Design Expo poster. Present supercharger prototype in Design Expo, and do formal presentation to Prof. Saitou and Prof. Skerlos	Complete
Final Paper	All	12/7/2006	12/11/2006	10%	US EPA: Andy Moskalik (734-214-4719)	Write final paper that presents concept selection as well as final designs for the prototype and final design.	Complete

APPENDIX G: FAST Diagram



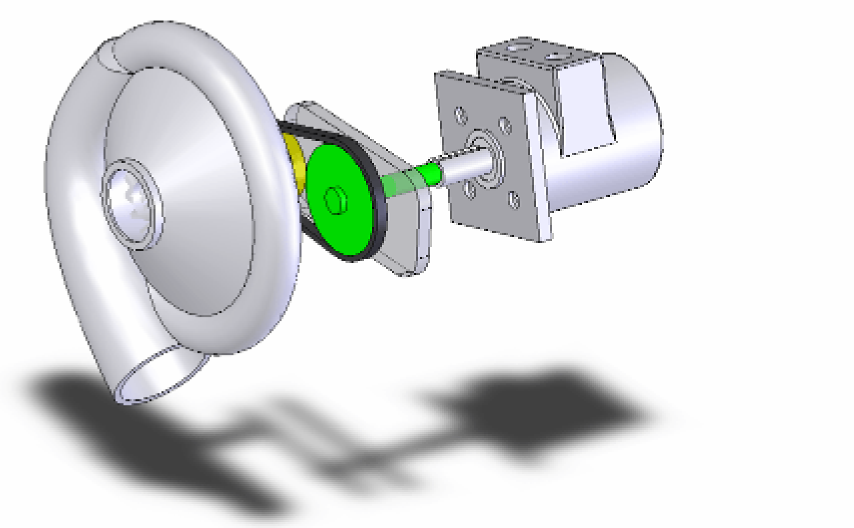
APPENDIX H: Morphological Chart

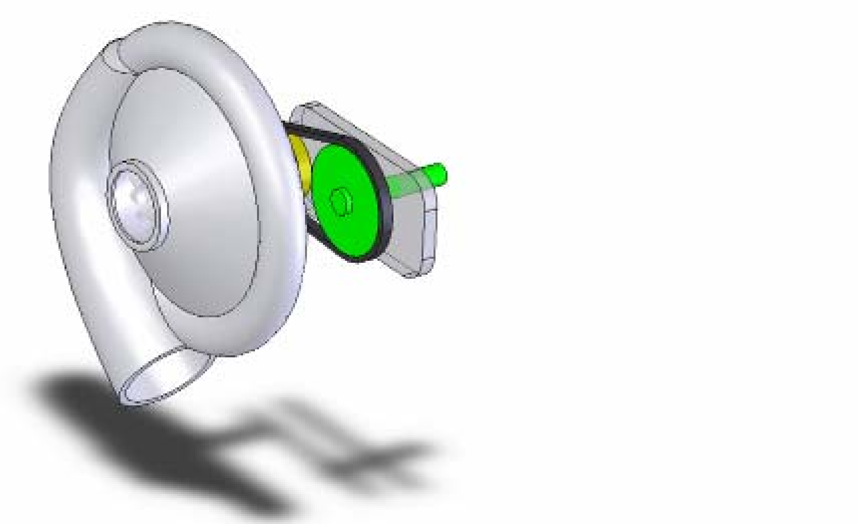
Function	Sub-function	Specific function	Concept 1	Concept 2	Concept 3	Concept 4
Transmits Power	Utilize hydraulic power					
		Extract hydraulic energy	hydraulic motor	hydraulic generator with electric motor	hydracharger paddle-wheel	
		Impart mechanical energy	shaft rotation	electrical energy driving electric motor		
	Rotate supercharger impeller	Implement gear/belt interface	gear system	belt system	shaft	
Pressurize air	Convert impeller motion	Optimize impeller geometry	radial vane type	francis vane type	mixed flow type	axial flow type
		Optimize casing geometry	screw type	centrifugal type		
		Control output	valve varying incoming pressure	valve varying outgoing pressure	valve varying inlet diameter	
Operation	User Interface	Boost activation	push button	switch	pedal	

APPENDIX I: Pugh Chart

Customer Requirements	Weight	CONCEPTS					
		BENCHMARK Centrifugal Compressor Paddlewheel Hydracharger Directly Driven Pressure Valve Control	**A** Centrifugal Compressor Hydraulic Motor Belt Driven Pressure Valve Control	**B** Centrifugal Compressor Hydraulic Generator - Electric Motor Gear Driven Outlet Valve Control	**C** Screw Type Compressor Paddlewheel Hydracharger Directly Driven Pressure Valve Control	**D** Screw Type Compressor Hydraulic Motor Gear Driven Outlet Valve Control	**E** Centrifugal Compressor Hydraulic Generator - Electric Motor Directly Driven Pressure Valve Control
Durable	0.75	S	+	-	S	-	+
Efficient	0.90	S	+	S	-	+	+
Quick to Power up	0.95	S	+	-	-	+	+
Meets Provided Pressure Profile	1.00	S	S	S	S	S	S
Meets Provided Flow Profile	1.00	S	S	S	S	S	S
Inexpensive	0.55	S	-	-	-	-	-
Recycles Hydraulic Fluid	1.00	S	+	+	S	+	+
Appearance	0.20	S	S	S	S	S	S
Use Existing Parts	0.80	S	+	+	S	+	+
Works in Set Hydraulic System	1.00	S	S	S	S	S	S
Easy to Manufacture	0.50	S	S	S	S	S	S
Easy to Repair	0.45	S	S	S	S	S	S
Simple User Interface	0.55	S	S	S	S	S	S
Low Weight	0.30	S	S	-	-	-	-
Easy to Install	0.65	S	+	-	S	+	-
Compact Size	0.65	S	+	+	S	+	+
∑+		**0.00**	**5.70**	**2.45**	**0.00**	**4.60**	**5.05**
∑-		**0.00**	**-0.55**	**-3.20**	**-2.70**	**-1.60**	**-1.50**
WEIGHTED TOTAL		**0.00**	**5.15**	**-0.75**	**-2.70**	**3.00**	**3.55**

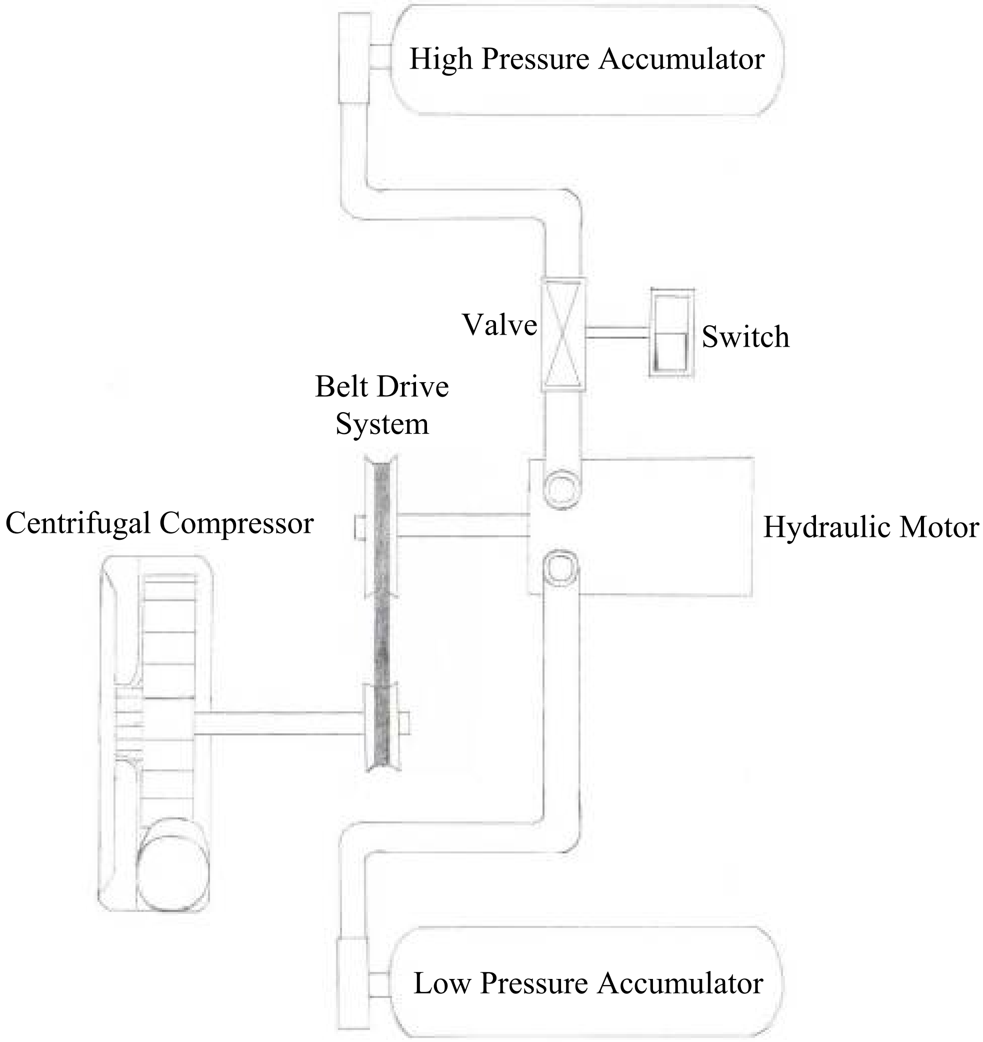
APPENDIX J: 3-D Model



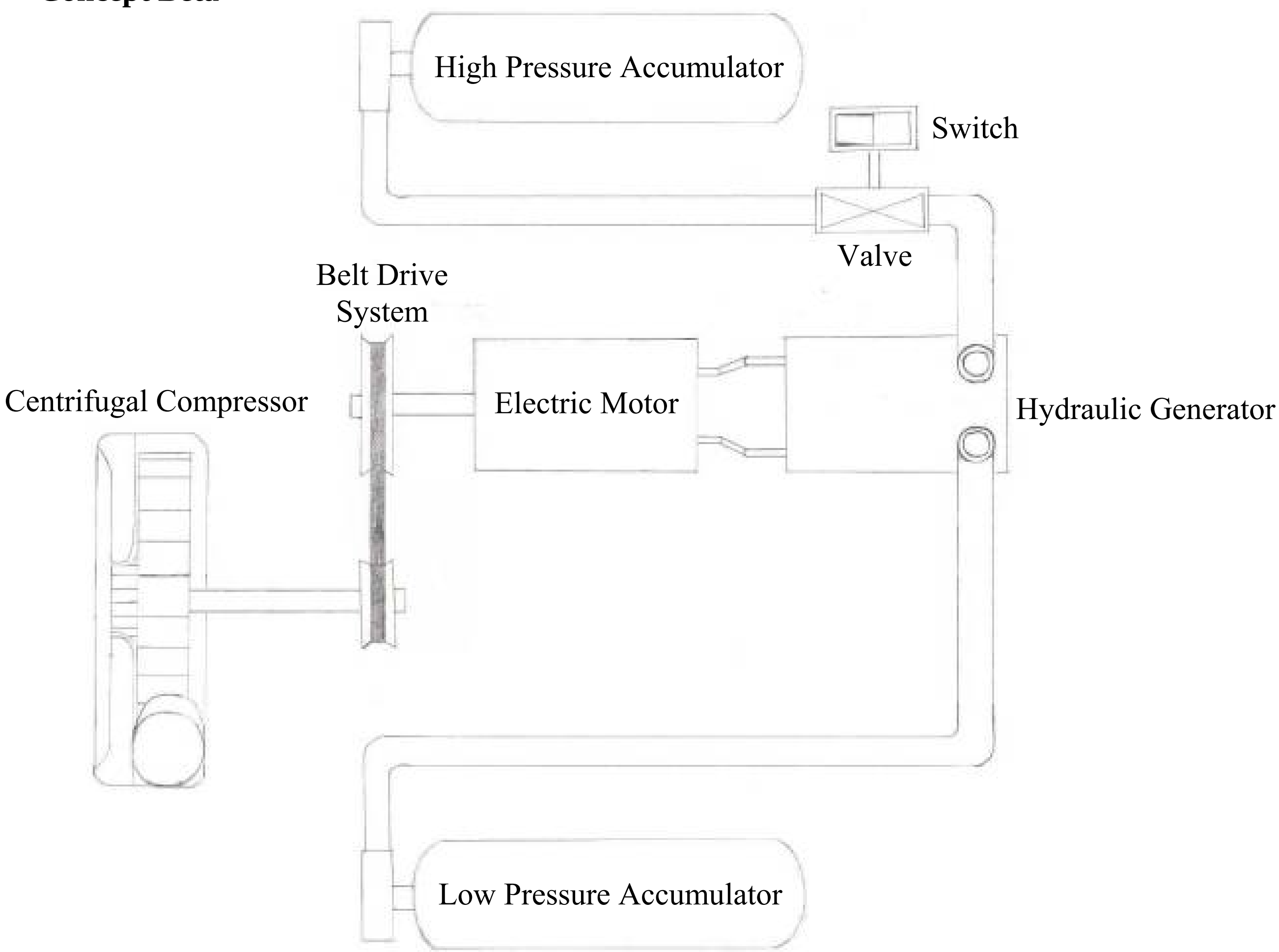


APPENDIX K: Concept Generation

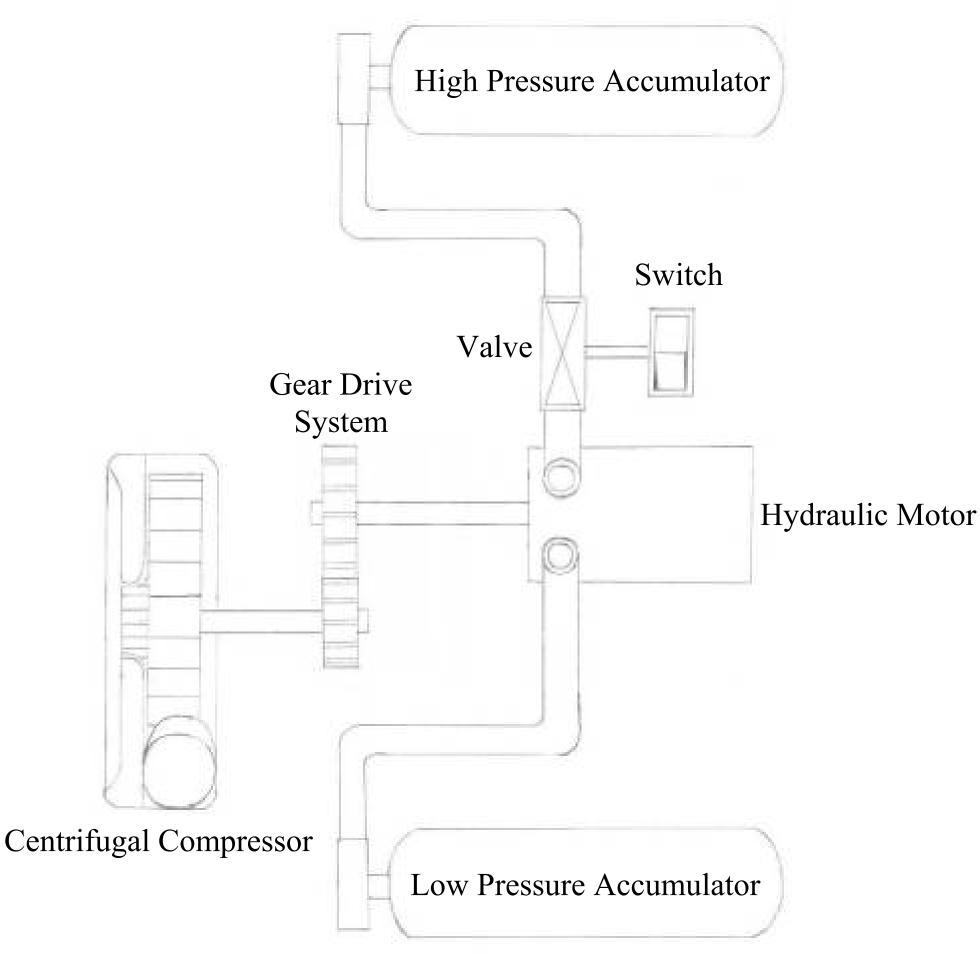
Concept Alpha



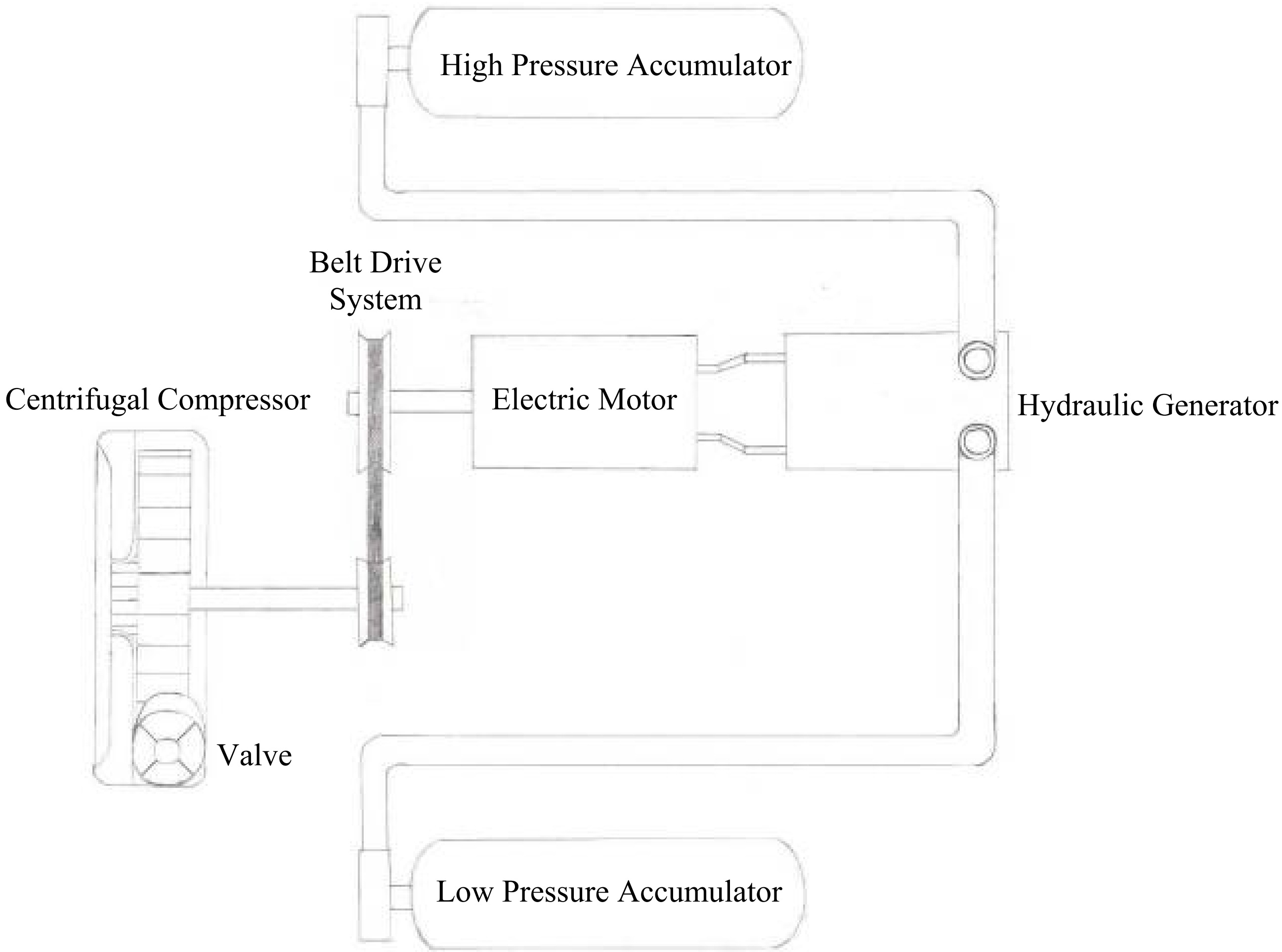
Concept Beta

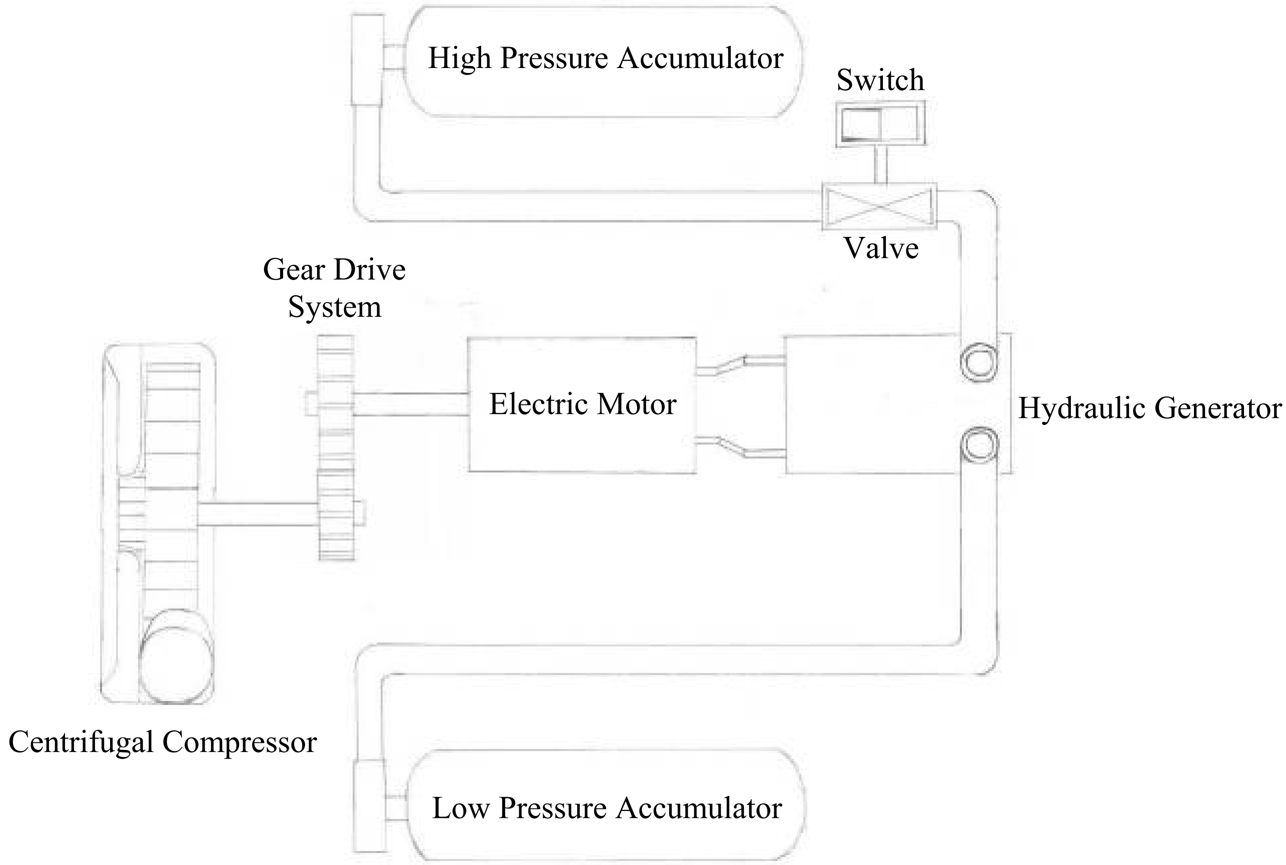


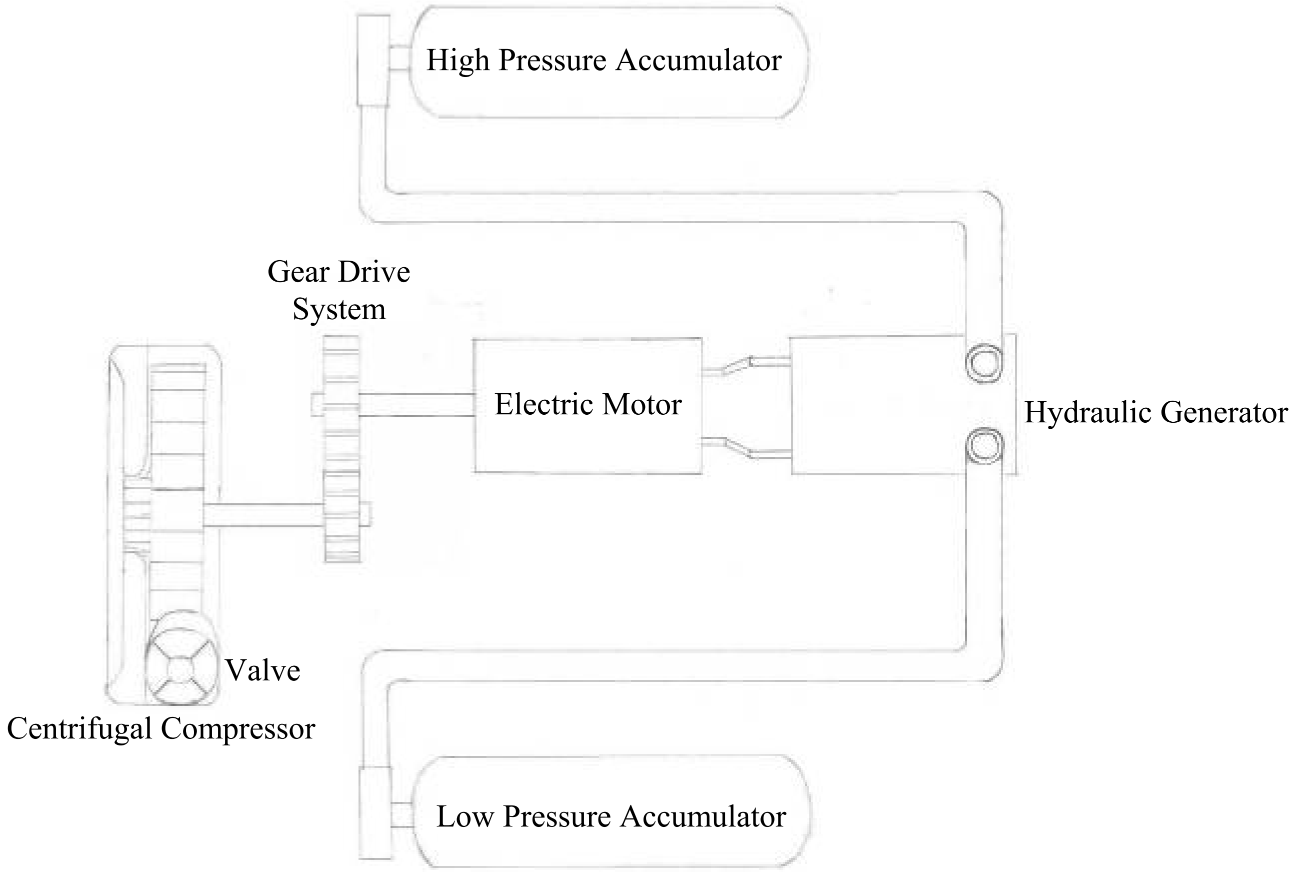
Concept Gamma



Additional Concepts




High Pressure Accumulator
Switch
Valve
Gear Drive
System
Electric Motor
Hydraulic Generator
Centrifugal Compressor
Low Pressure Accumulator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Gear Drive
System
Electric Motor
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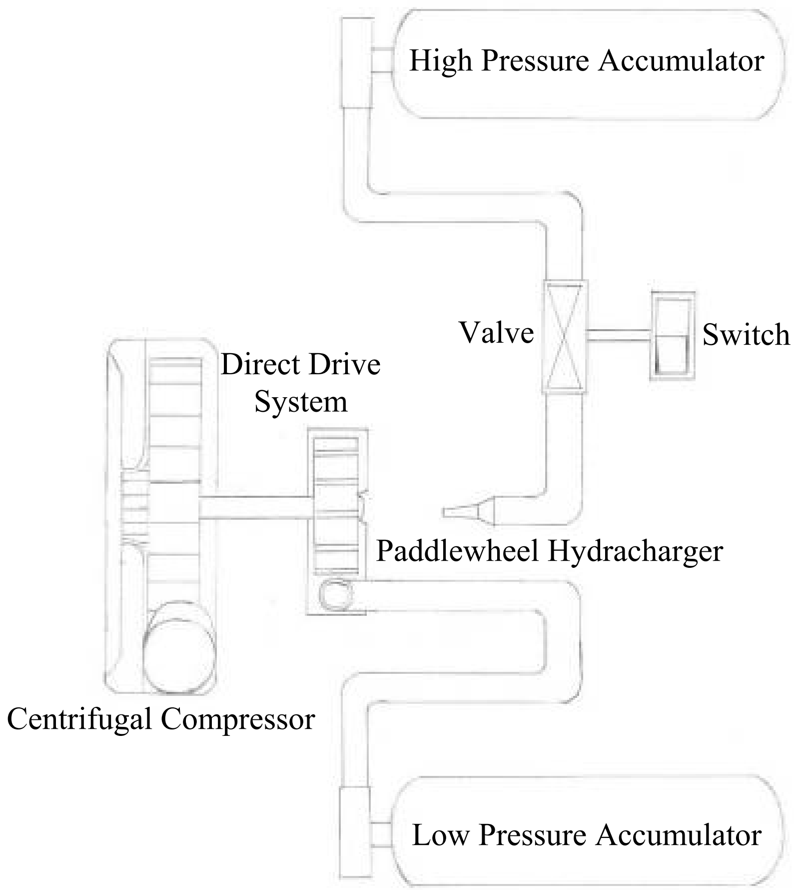
Valve
Centrifugal Compressor
Low Pressure Accumulator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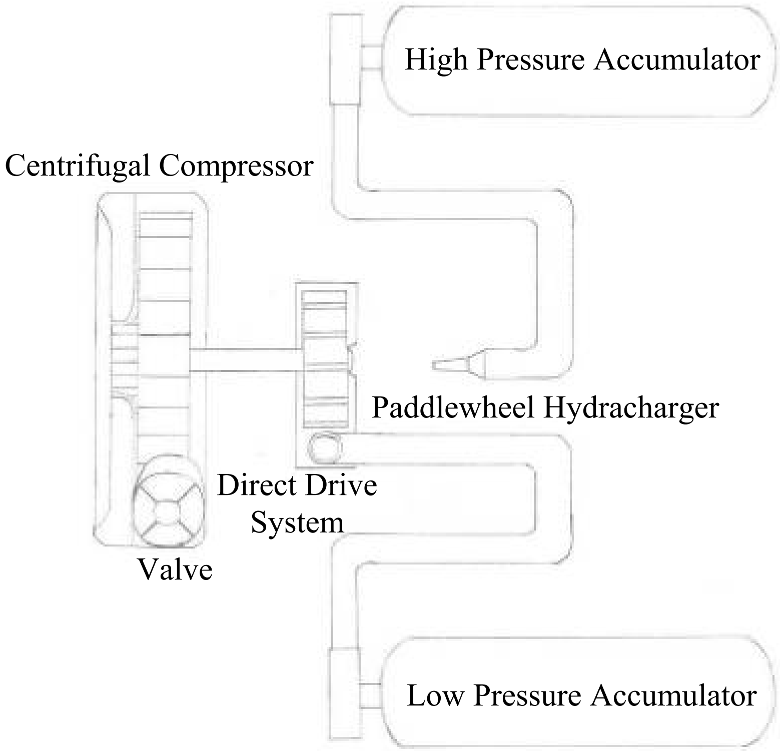
Switch
Direct Drive
System
Paddlewheel Hydracharger
Centrifugal Compressor
Low Pressure Accumulator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Centrifugal Compressor
Paddlewheel Hydracharger
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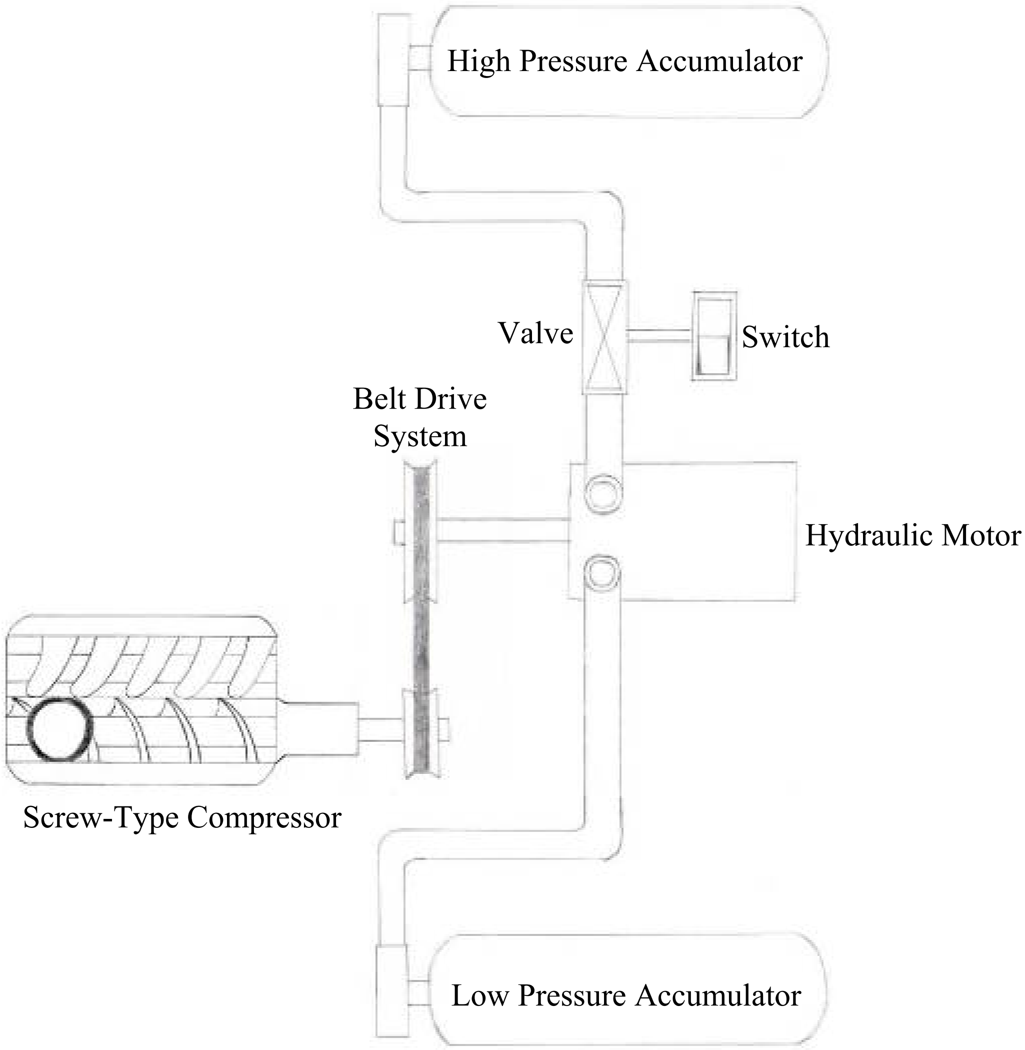
Valve
Low Pressure Accumulator


High Pressure Accumulator
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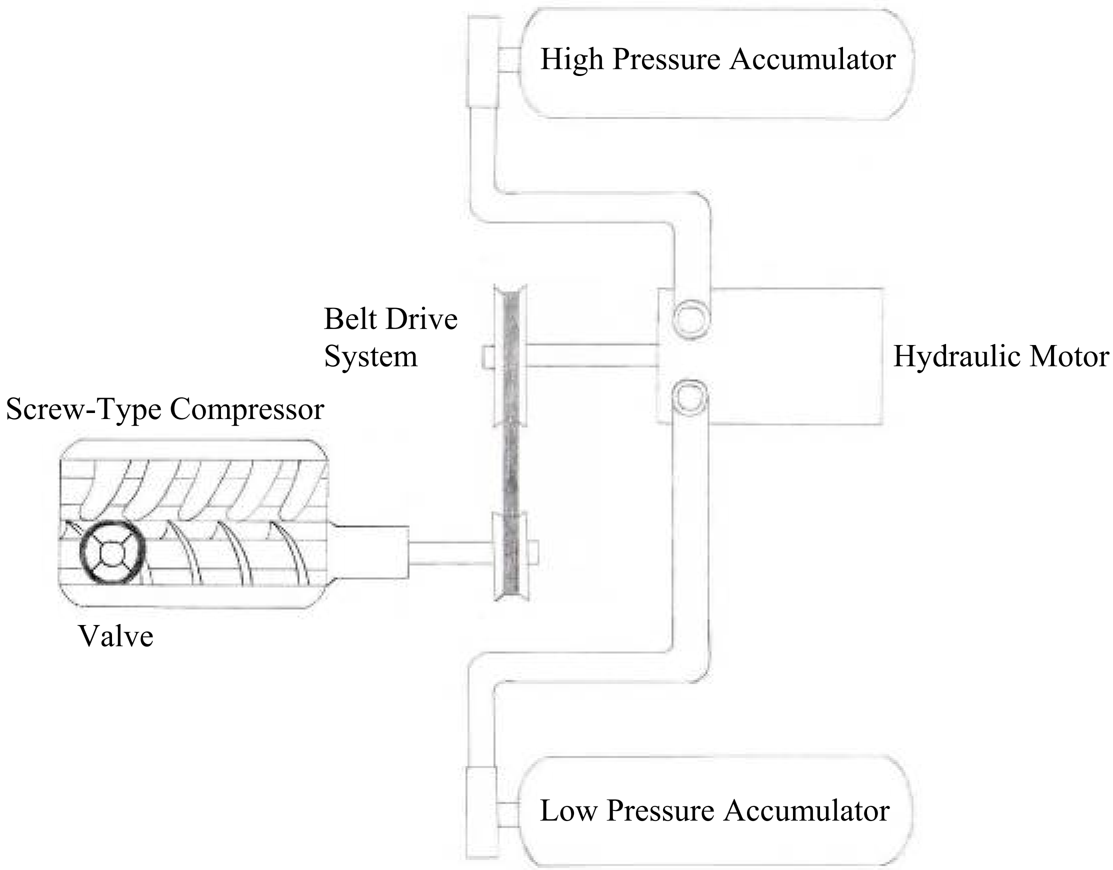
Belt Drive
System
Hydraulic Motor
Screw-Type Compressor
Low Pressure Accumulator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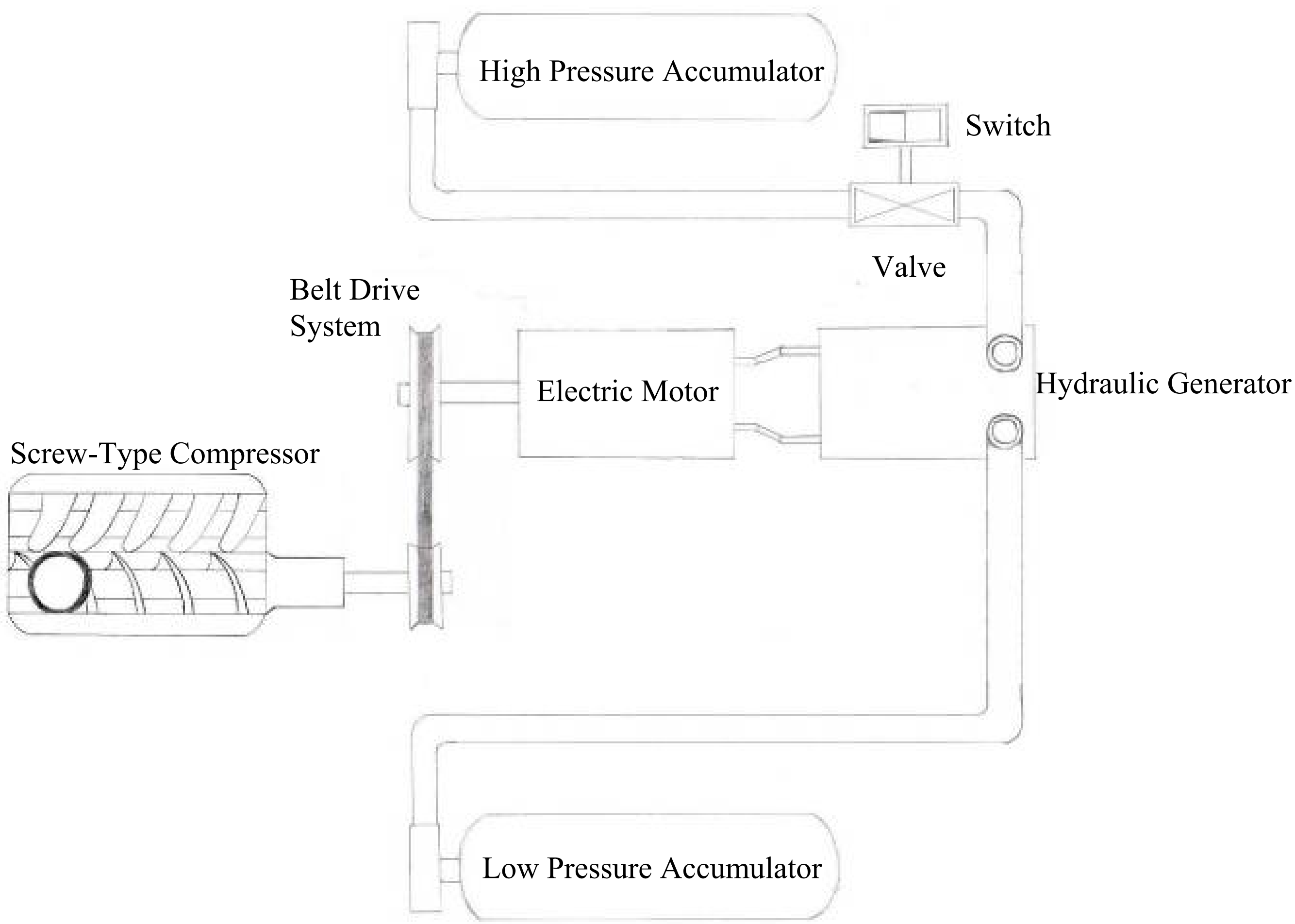
Valve
Low Pressure Accumulator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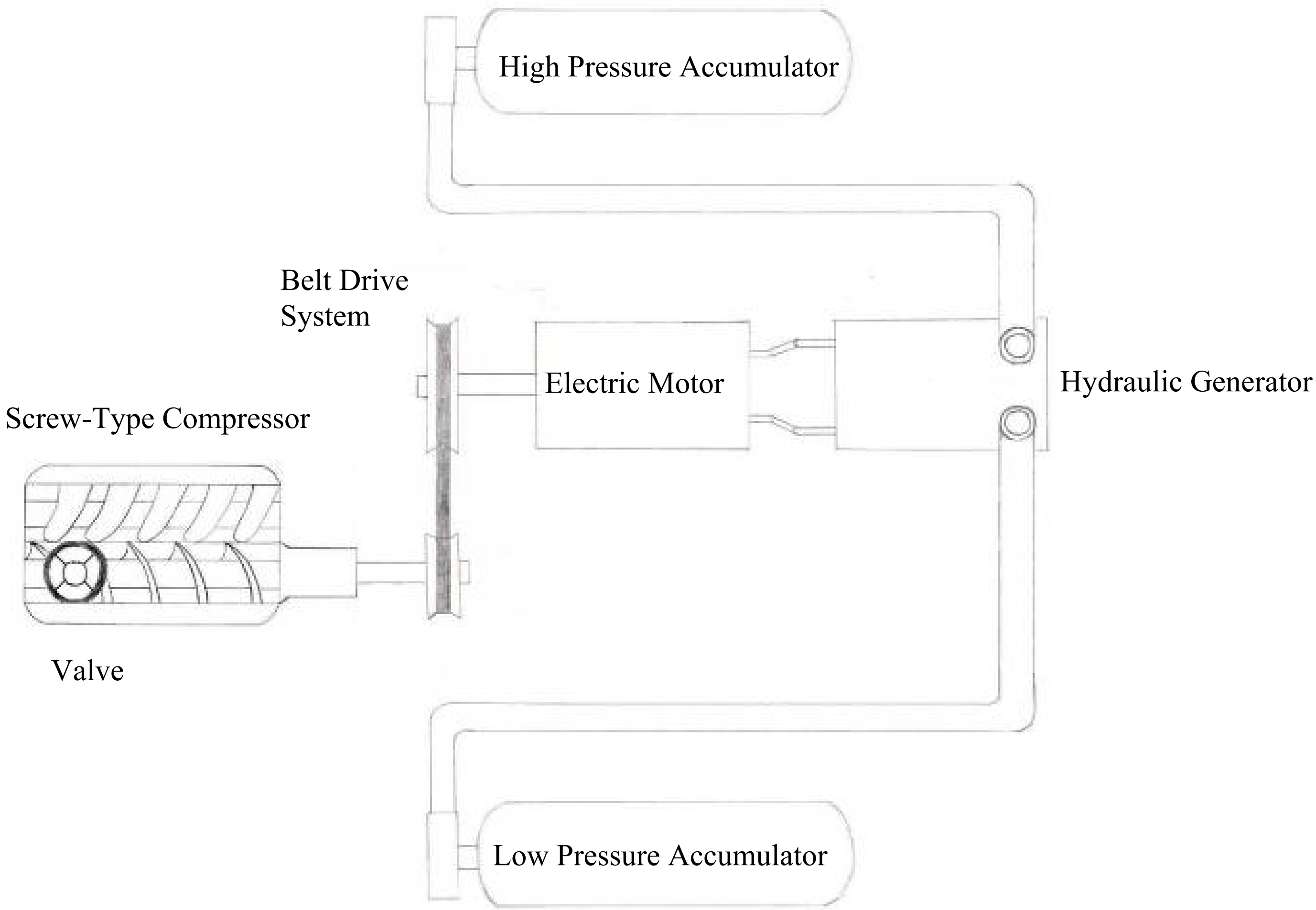
Screw-Type Compressor
Low Pressure Accumulator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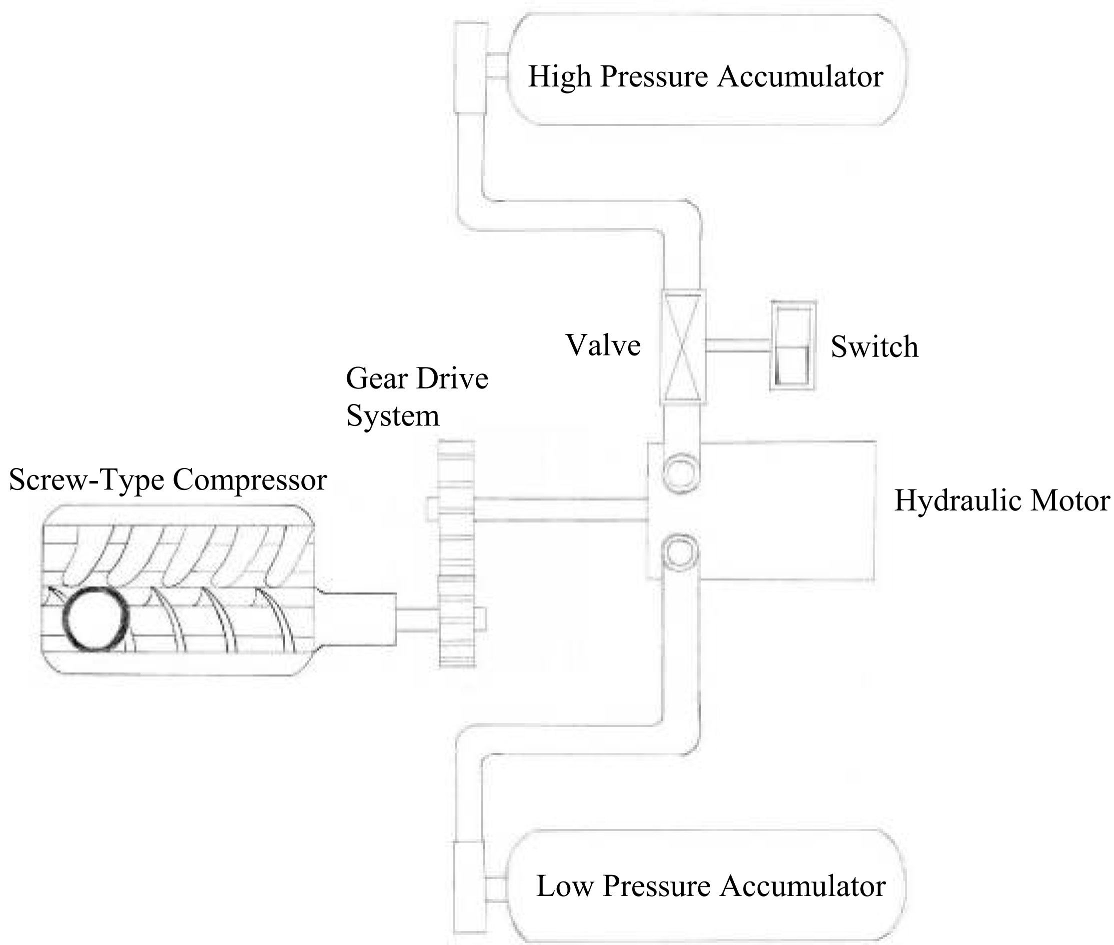
Belt Drive
System
Electric Motor
Hydraulic Generator
Screw-Type Compressor
Valve
Low Pressure Accumulator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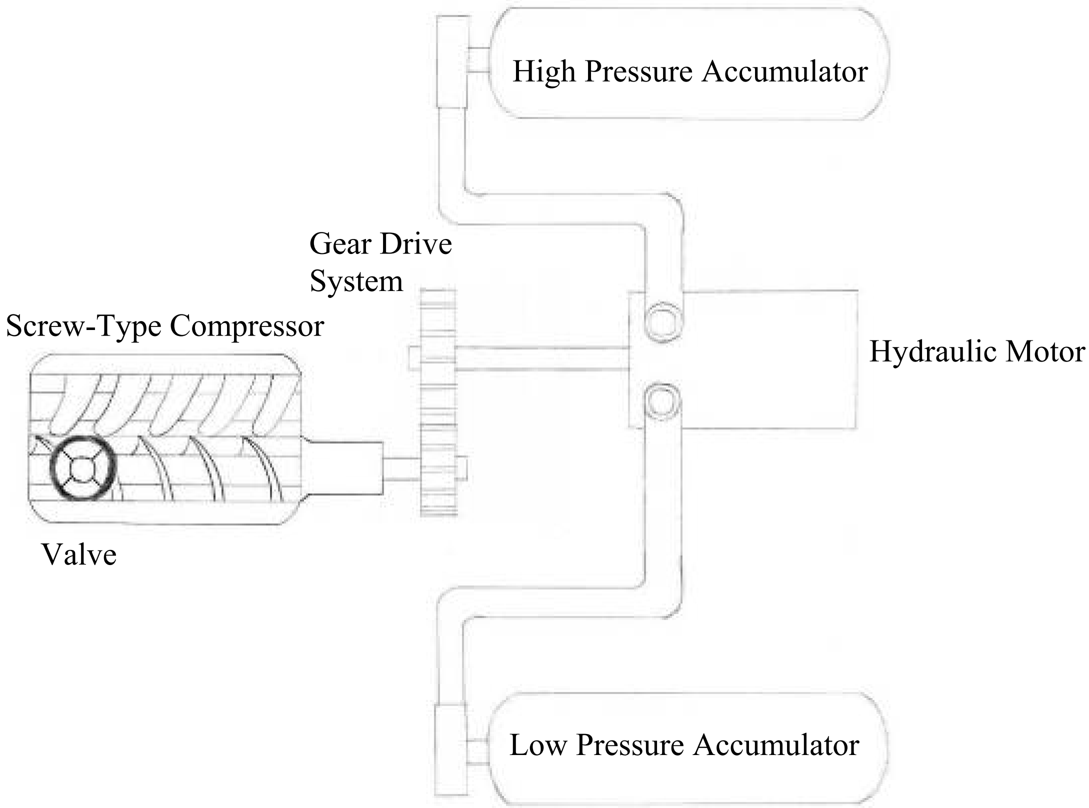
Valve
Switch
Gear Drive
System
Screw-Type Compressor
Hydraulic Motor
Low Pressure Accumulator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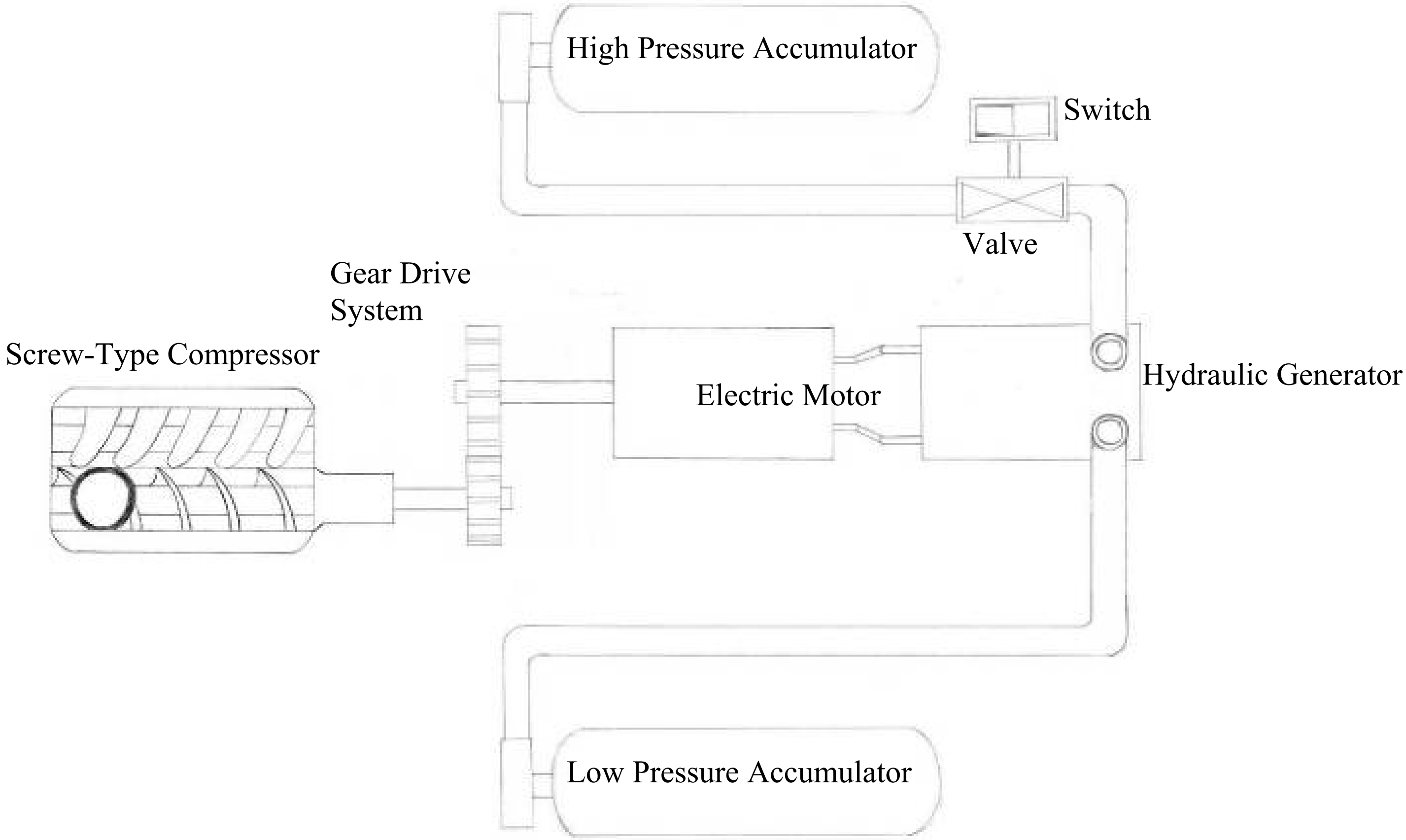
Gear Drive
System
Screw-Type Compressor
Hydraulic Motor
Valve
Low Pressure Accumulator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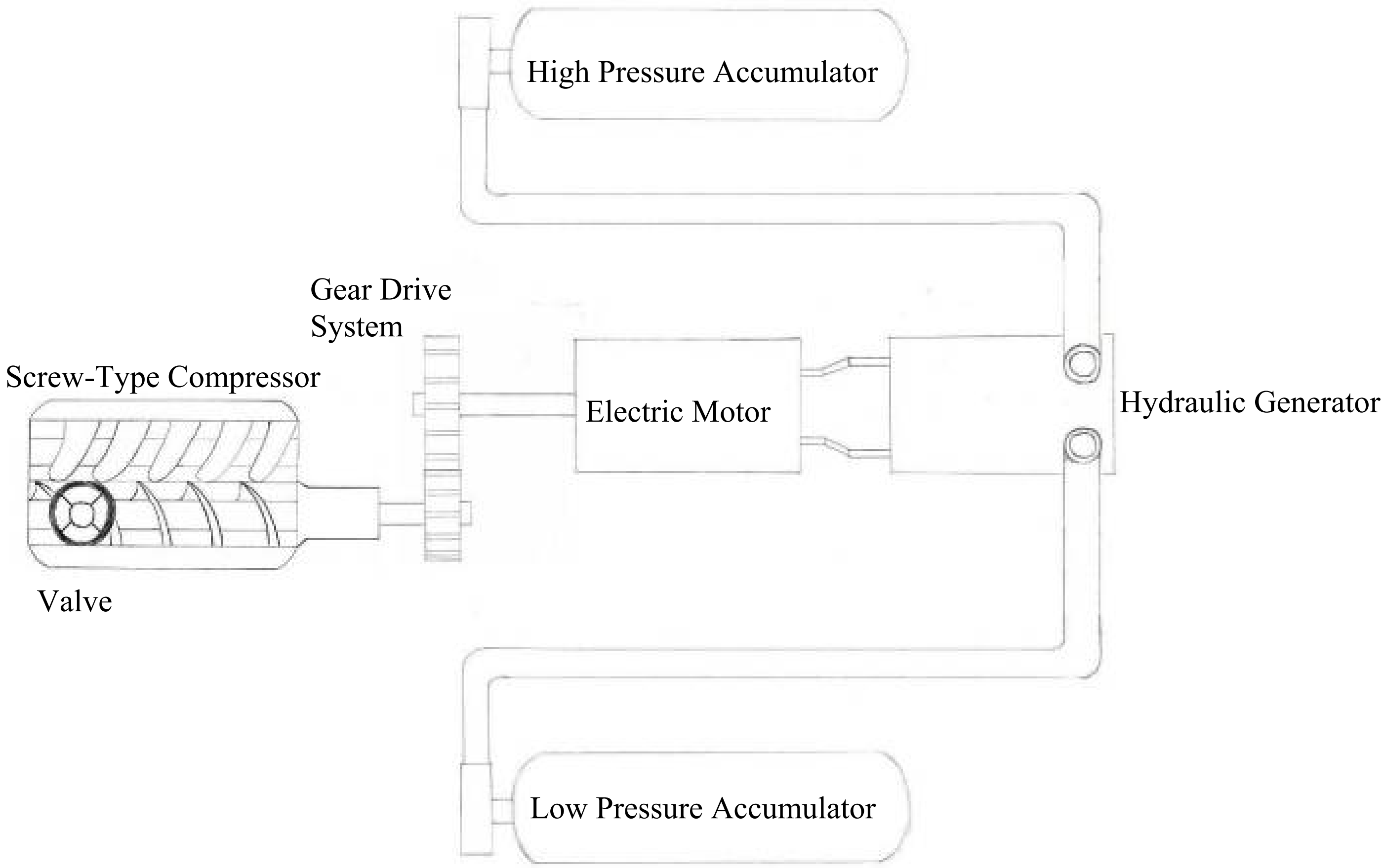
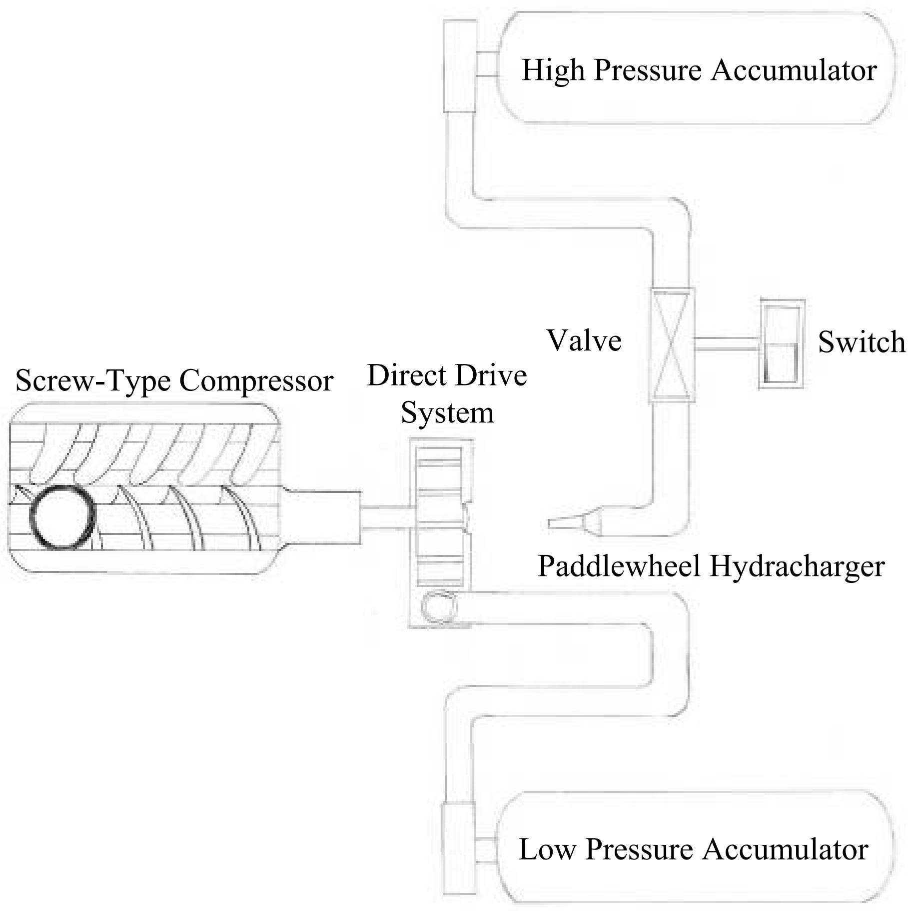
Switch
Valve
Gear Drive
System
Screw-Type Compressor
Electric Motor
Hydraulic Generator
Low Pressure Accumulator


High Pressure Accumulator
Gear Drive
System
Screw-Type Compressor
Electric Motor
Hydraulic Generator
Valve
Low Pressure Accumulator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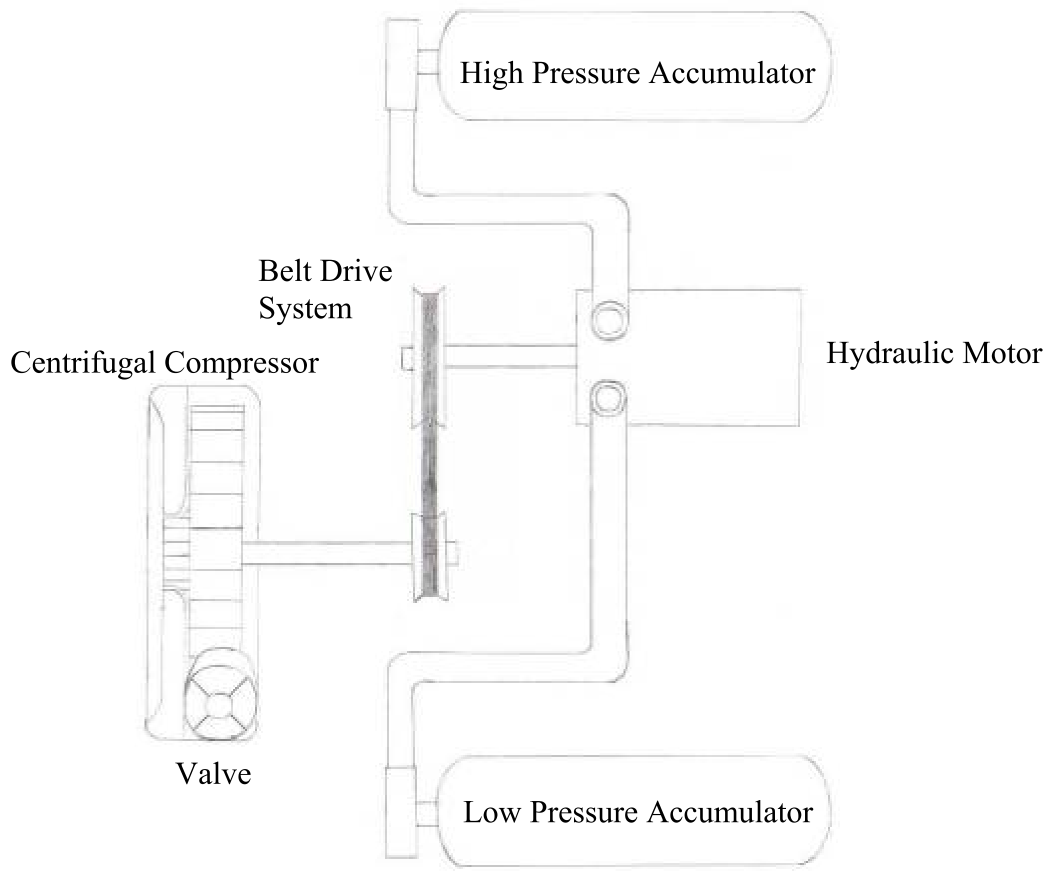
Valve
Switch
Screw-Type Compressor
Direct Drive
System
Paddlewheel Hydracharger
Low Pressure Accumulator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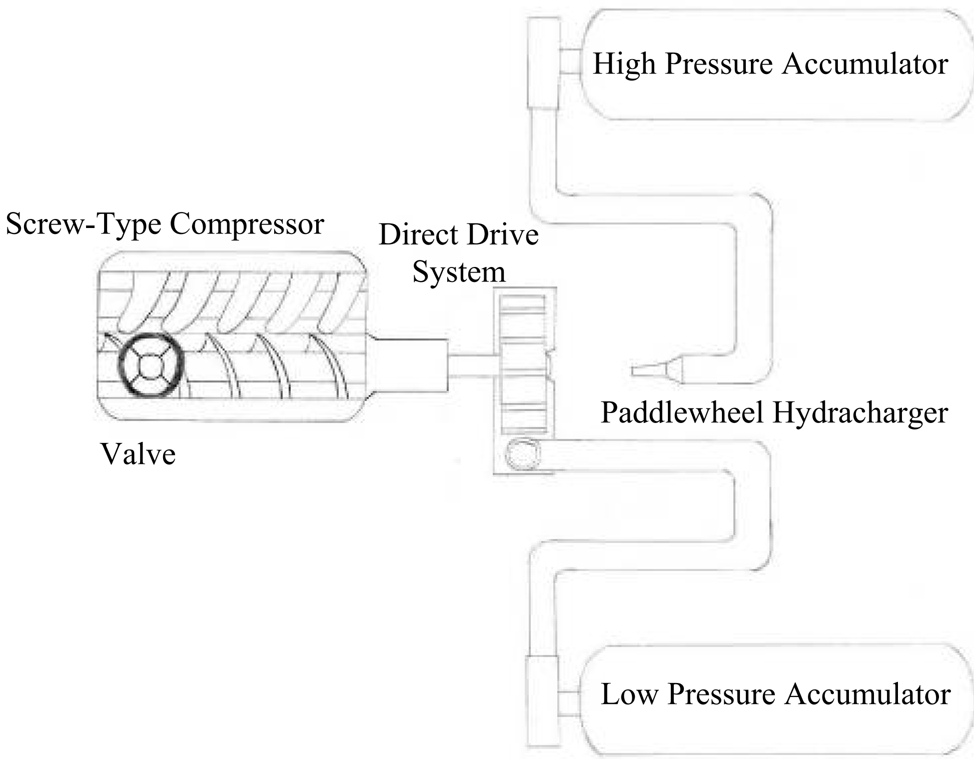
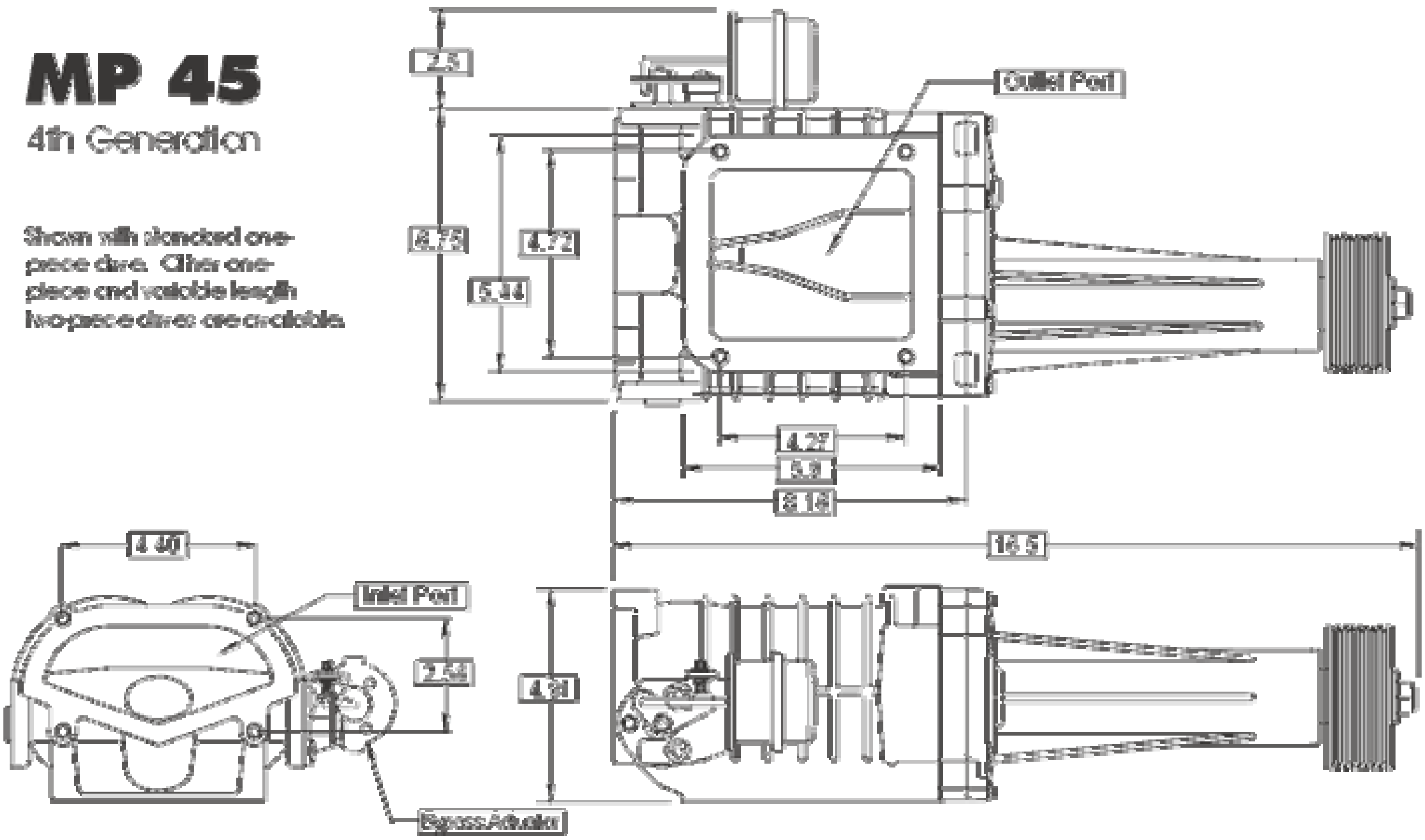
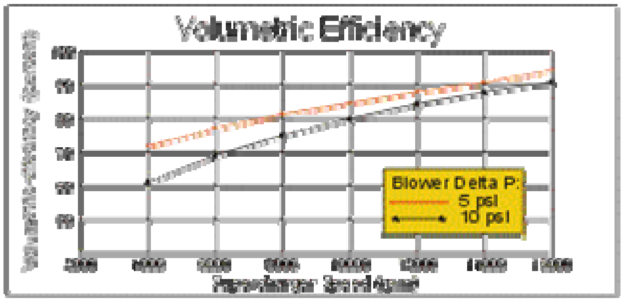
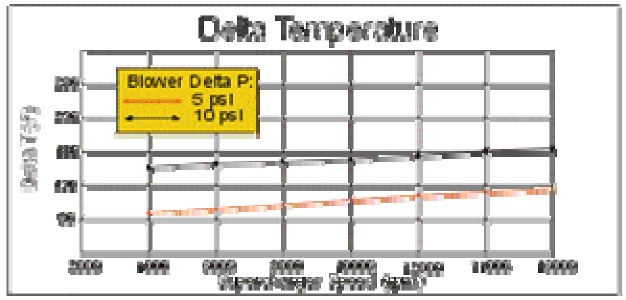
Belt Drive
System
Centrifugal Compressor
Hydraulic Motor
Valve
Low Pressure Accumulator

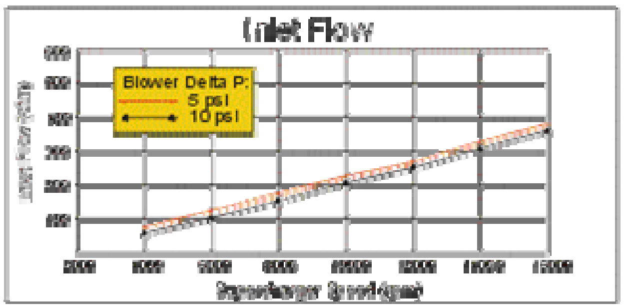

High Pressure Accumulator
Screw-Type Compressor
Direct Drive
System
Paddlewheel Hydracharger
Valve
Low Pressure Accumulator

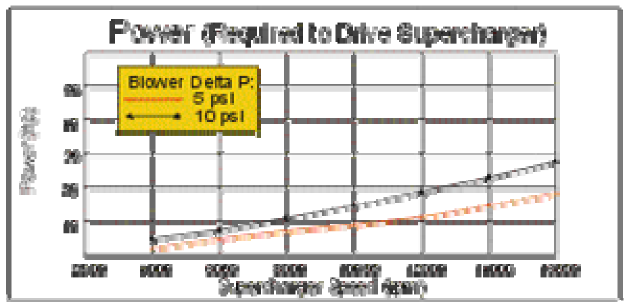
APPENDIX L: Eaton MP45 Supercharger Specifications



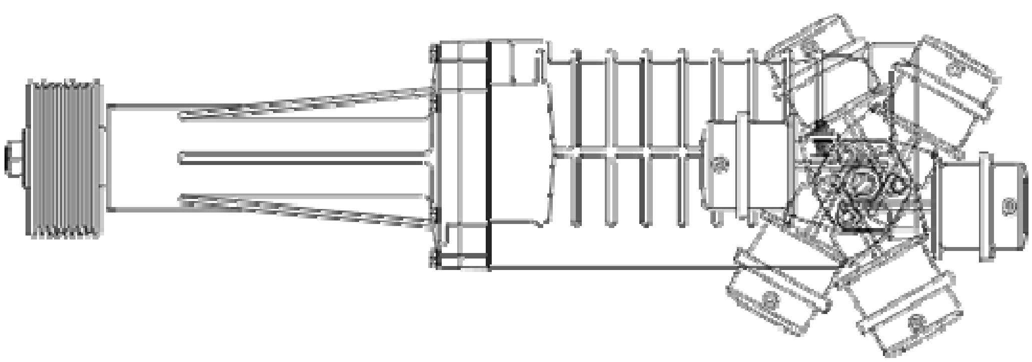








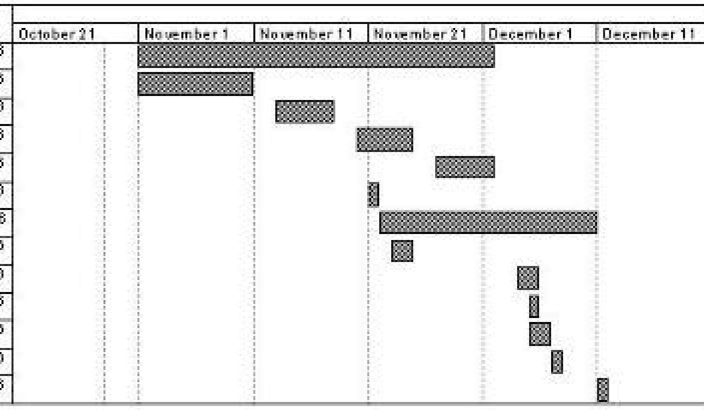
4th Generation Actuator Options



Actuator can mount in any one of 12 possible positions with 6 on each side.

APPENDIX M: Initial Manufacturing Plan

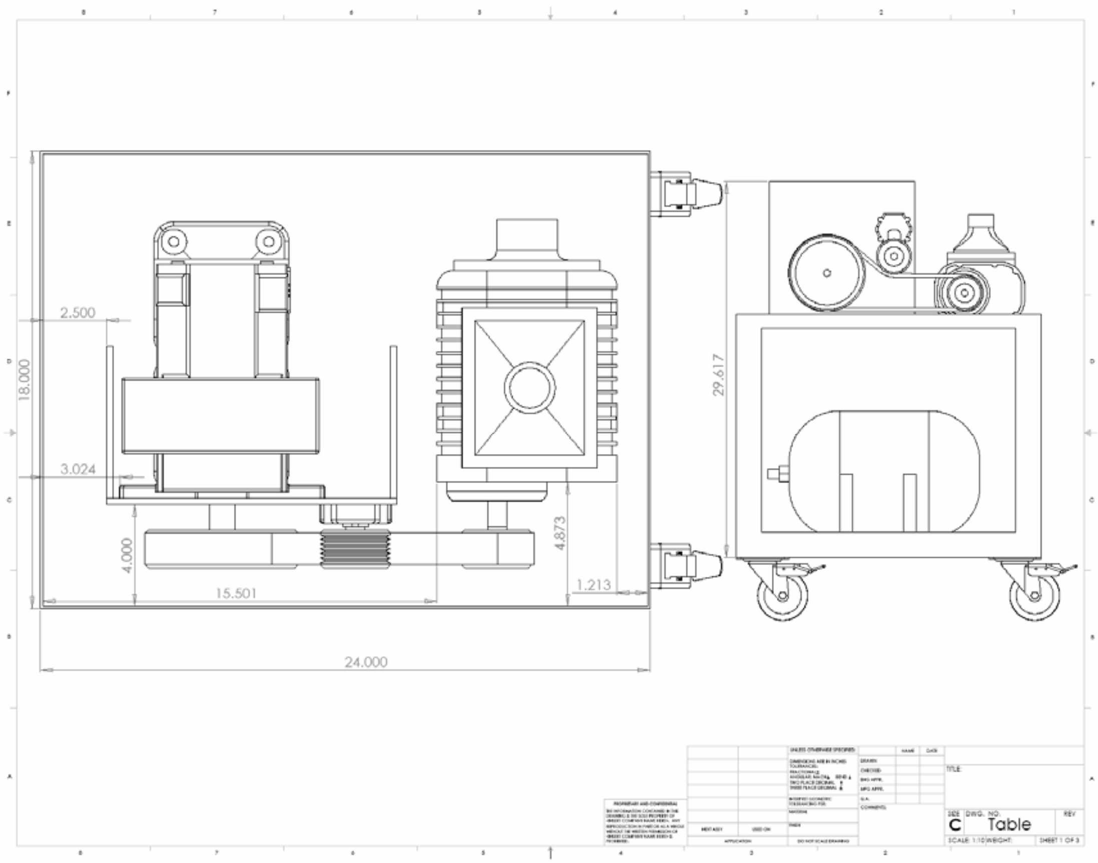
ID	Task Name	Duration	Start	Finish
1	Beta Prototype Fabrication	23 days	Wed 11/1/06	Fri 12/1/06
2	Manufacture Belt/Pulley Drive	8 days	Wed 11/1/06	Fri 11/10/06
3	Manufacture Tensioner for Belt	5 days	Mon 11/13/06	Fri 11/17/06
4	Manufacture Compressor Manifold / Belt Drive Fixture	5 days	Mon 11/20/06	Fri 11/24/06
5	Manufacture Boost Control Mechanism	5 days	Mon 11/27/06	Fri 12/1/06
6	Design Review #4	1 day	Tue 11/21/06	Tue 11/21/06
7	Prepare Final Report	13 days	Wed 11/22/06	Sun 12/10/06
8	Thanksgiving	2 days	Thu 11/23/06	Fri 11/24/06
9	Assemble and Finalize Beta Prototype for Testing/Expo	2 days	Mon 12/4/06	Tue 12/5/06
10	Discussion: Expo Logistics and Section Evaluation	1 day	Tue 12/5/06	Tue 12/5/06
11	Test Beta Prototype	2 days	Tue 12/5/06	Wed 12/6/06
12	Design Expo	1 day	Thu 12/7/06	Thu 12/7/06
13	Final Report Due	1 day	Mon 12/11/06	Mon 12/11/06

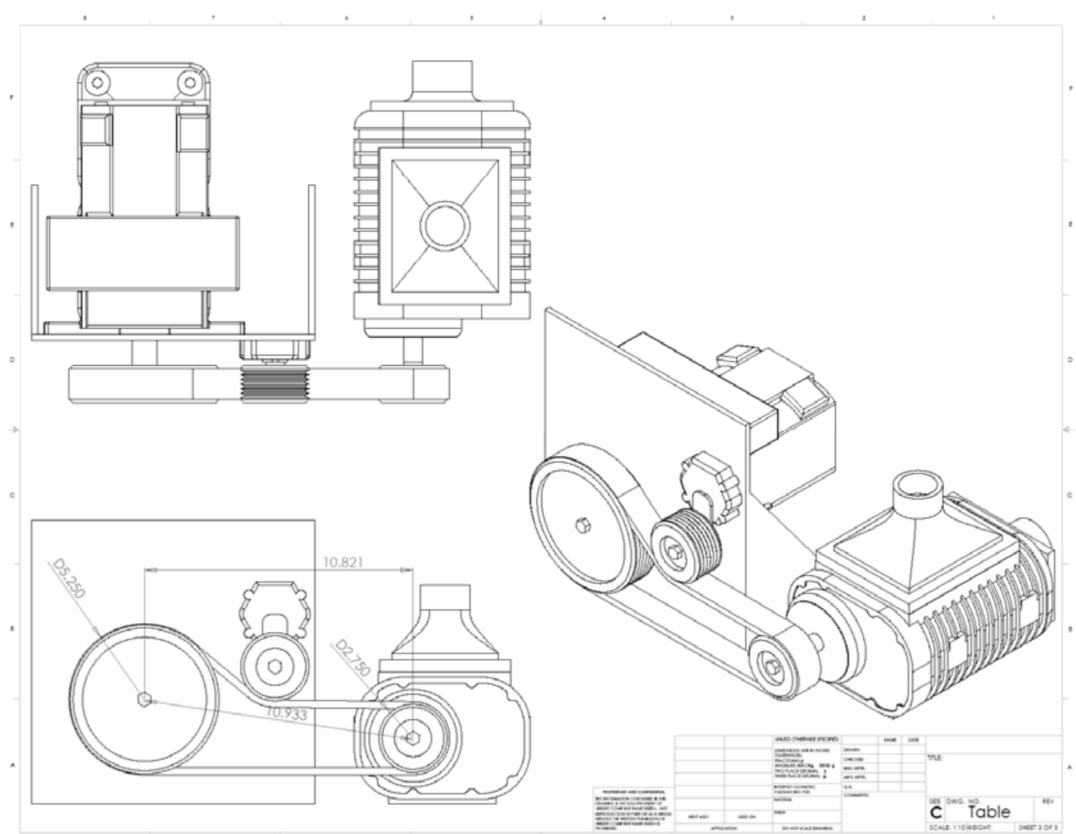


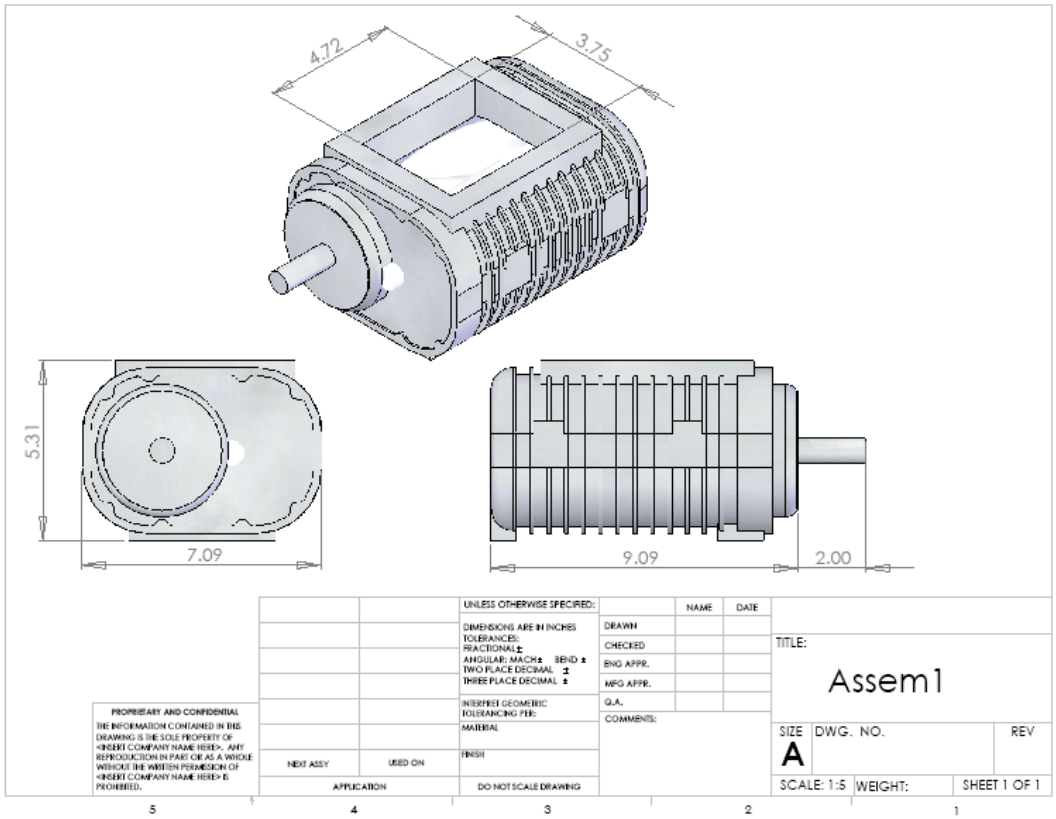
Project: Initial Manufacturing Plan
Date: Sun 10/29/06

Page 1

APPENDIX N: Final Design Layout Drawings



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THE INFORMATION CONTAINED IN THIS DRAWING IS THE SOLE PROPERTY OF <INSERT COMPANY NAME HERE>. ANY REPRODUCTION IN PART OR AS A WHOLE WITHOUT THE WRITTEN PERMISSION OF <INSERT COMPANY NAME HERE> IS PROHIBITED.
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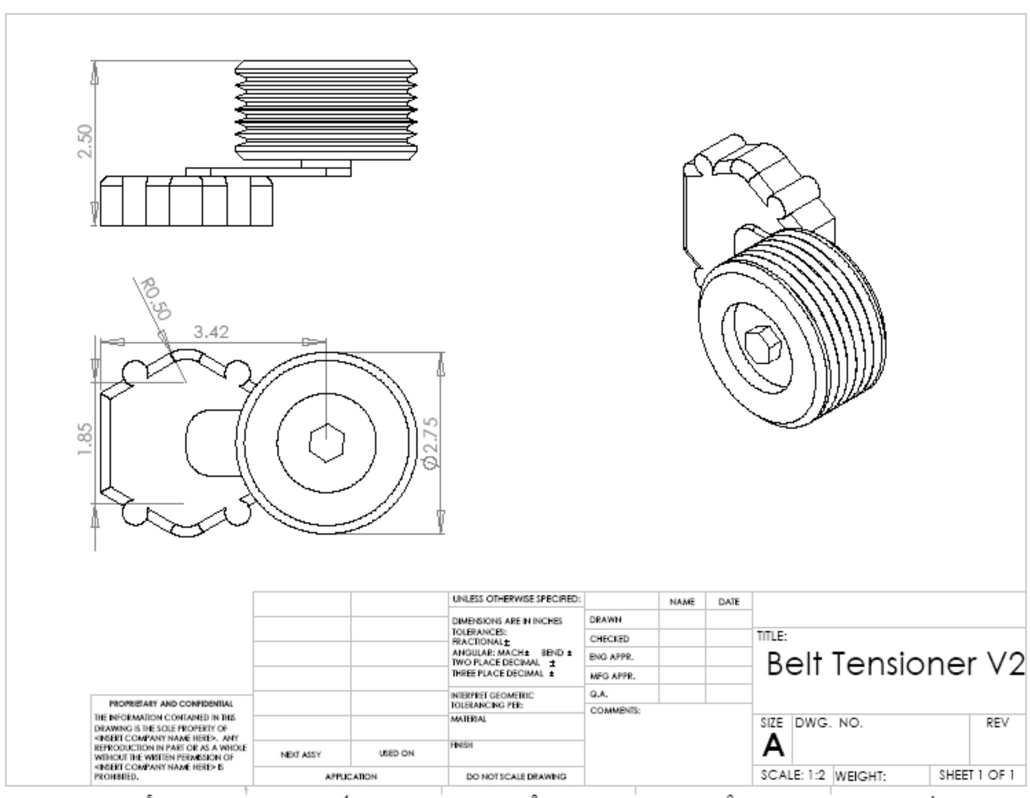
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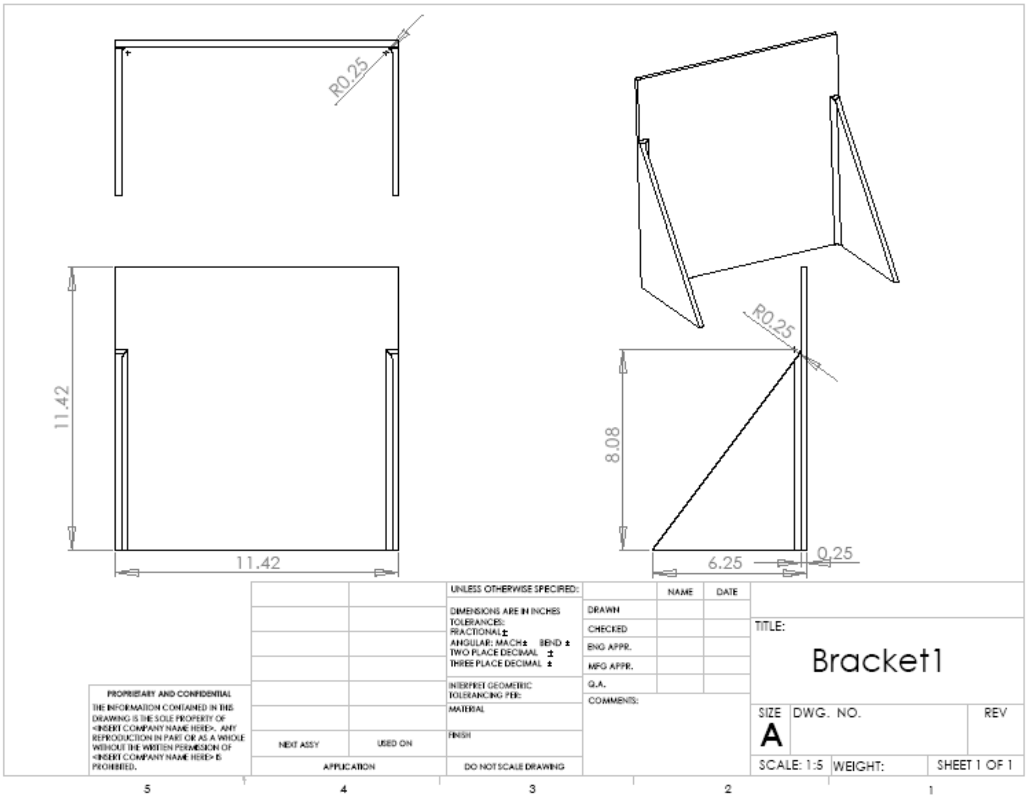
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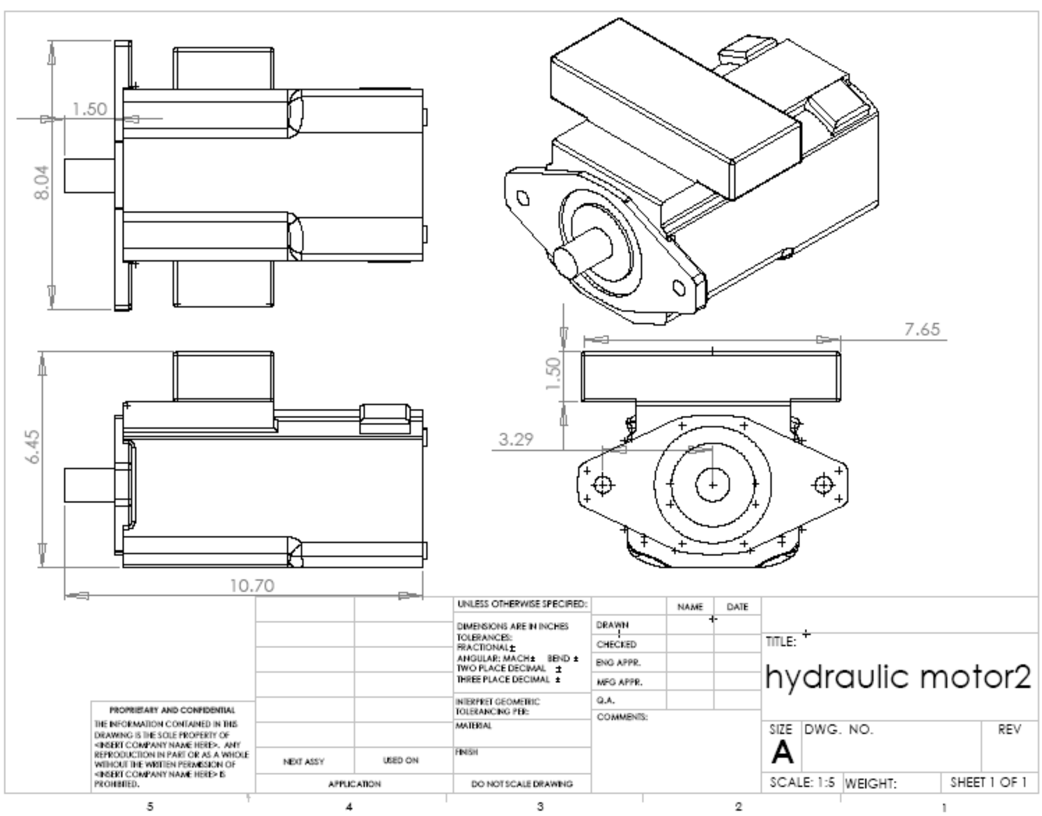
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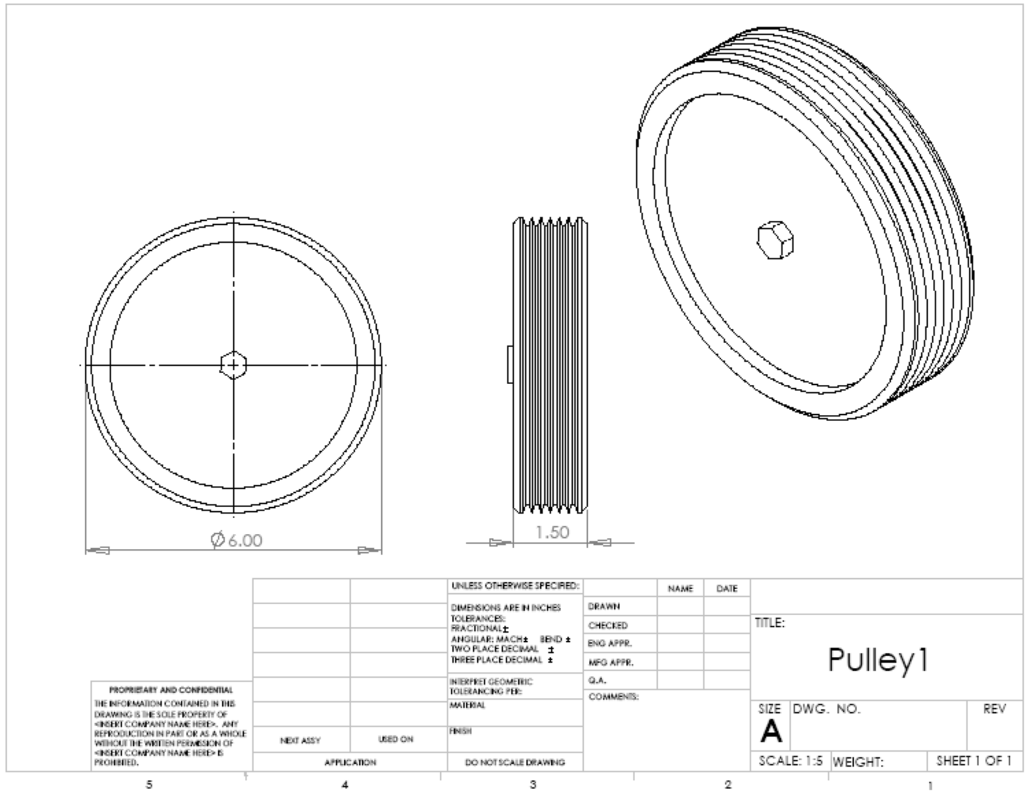
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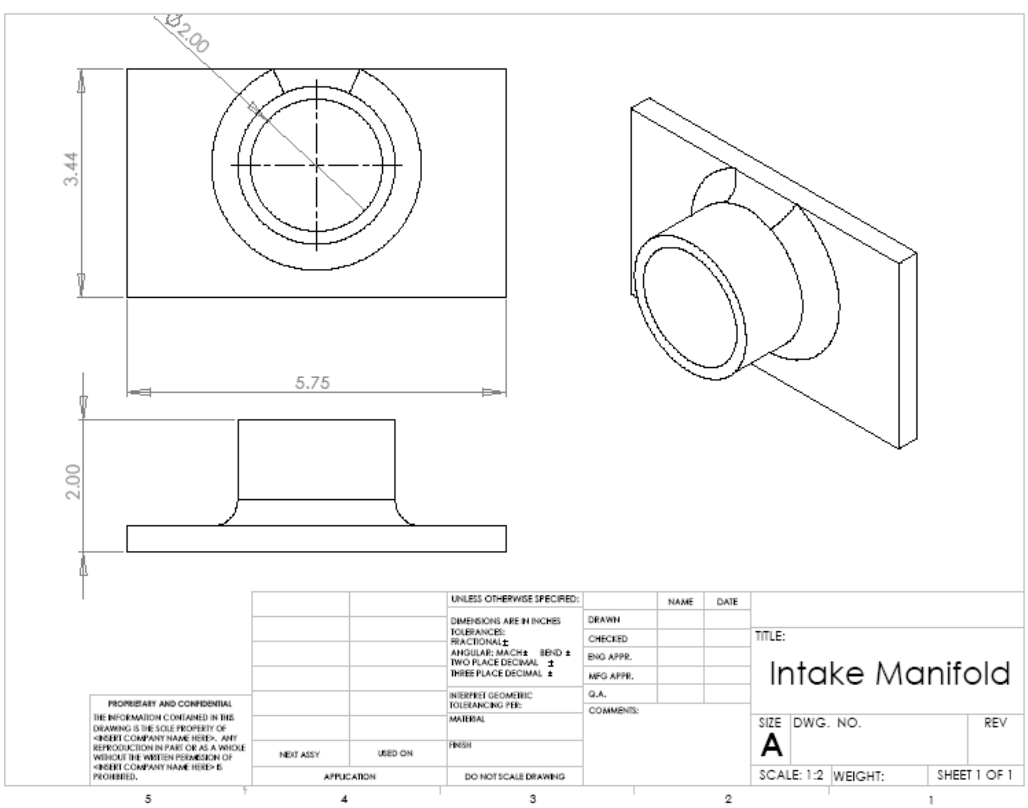
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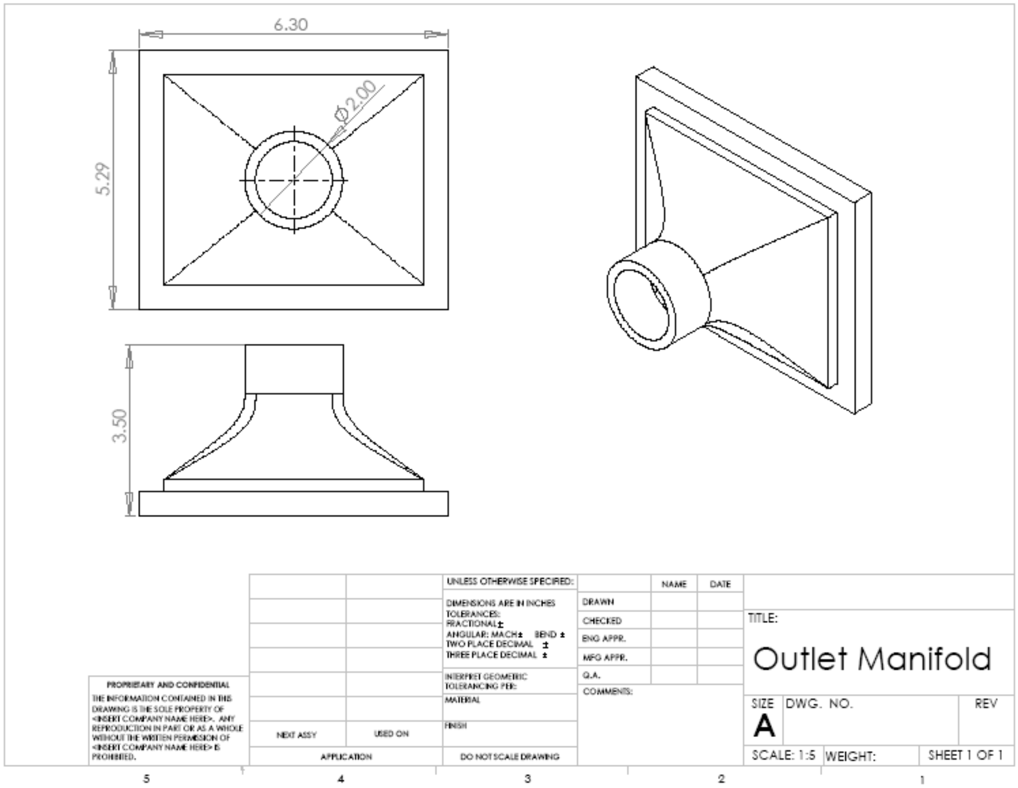

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APPENDIX O: DFMEA

DESIGN FAILURE MODE AND EFFECTS ANALYSIS (DFMEA)

Product Name: ____________

____________ System

____________ Subsystem Name: ____________

____________ Component

Prepared By Development Team: ____________

DFMEA Origination Date: ____________

Revision Prepared By: ____________

Rev. Date: ____________

Row #	Item/Function	Potential Failure Mode	Potential Effect(s) of Failure	Severity (S)	Potential Cause(s)/ Mechanism(s) of Failure	Occurrence (O)	Current Design Controls/ Tests	Detection (D)	Recommended Actions	RPN	Recommended Actions	Responsibility & Target Completion Date	Action Results: Actions Taken	S	O	D	RPN
	COMPRESSOR																0
1	compressor: outlet diameter provides adequate	ID too small for flow path	decrease in boost pressure	7	design error, output exceeds compressor capacity	1		3		21							0
2	compressor: inlet diameter provides adequate flow path	ID too small for flow path	decrease in boost pressure	7	design error, input exceeds compressor capacity	1		3		21							0
3	compressor: screw geometry produces boost	screw geometry deformed	decrease in boost pressure	7	supplier design error, incorrect selection of material for RPMs	3		3		63							0
4	compressor: screw material durable for reasonable cycles	screw geometry deteriorates	decrease in boost pressure	7	supplier design error	1		7		49							0
5	compressor: screw material/geometry limits frictional losses	friction in screw	decrease in boost pressure	7	supplier design error	1		7		49							0
6	compressor: screw tolerance limits frictional	friction in screw	decrease in boost pressure	7	supplier design error, incorrect selection of finish for screw	1		7		49							0
7	compressor: shaft material/geometry limits frictional losses	friction in shaft	decrease in boost pressure	7	supplier design error, incorrect selection of materials for shaft	1		7		49							0
8	compressor: shaft tolerance limits frictional losses	friction in shaft	decrease in boost pressure	7	supplier design error, incorrect selection of finish for shaft	1		7		49							0
9	compressor: shaft material durable for reasonable cycles of use	shaft material yields	complete loss of boost pressure	9	supplier design error, incorrect selection of materials for shaft	1		7		63							0
10	compressor: outlet diameter provides pressure boost to intake manifold of engine	ID too small for flow path	decrease in boost pressure	7	design error, output exceeds compressor capacity	3		3		63							0
11	compressor: outlet tube mates with intake manifold, with no leakage of air providing boost	air boost pressure leaks	decrease in boost pressure	7	design error, incorrect finish on mating parts	3		3		63							0
12	compressor: interior material strength/hardness withstands hydraulic fluid flow/pressure	body material ruptures	complete loss of boost pressure	9	supplier design error, incorrect selection of materials for strength property	1		7		63							0
	BELT/PULLEY																0
13	belt/pulley: belt material durable for reasonable cycles of use	edge cord failure	complete loss of boost pressure	9	incorrect selection of materials for strength property, excessive tension, drive pulley misalignment, improper pulley support	5		3		135							0

Row #	Item/Function	Potential Failure Mode	Potential Effect(s) of Failure	Severity (S)	Potential Cause(s)/ Mechanism(s) of Failure	Occurrence (O)	Current Design Controls/ Tests	Detection (D)	Recommended Actions	RPN	Recommended Actions	Responsibility & Target Completion Date	Action Results				
													Actions Taken				R
14	belt/pulley: belt material durable for reasonable cycles of use	sporadic rib cracking	complete loss of boost pressure	9	incorrect selection of pulley diameter, excessive heat conditions, excessive belt thickness, belt overcure, excessive	5		3		135							0
15	belt/pulley: belt material durable for reasonable cycles of use	rib chunking	complete loss of boost pressure	9	progression from rib cracking, debris in pulley grooves, installation damage, excessive drive loads	5		3		135							
16	belt/pulley: belt material does not slip	belt material becomes oily	decrease in boost pressure	7	leak of oil/hydraulic fluid onto belt reducing frictional characteristics	5		3		105							0
17	belt/pulley: belt material does not slip	belt material becomes worn	decrease in boost pressure	7	incorrect selection of materials for strength property, drive misalignment, mismatch of belt and pulley groove widths, use of soft rubber compounds	5		3		105							0
18	belt/pulley: belt material does not slip	failure of belt to carry load	decrease in boost pressure	7	insufficient belt tension, belt hardened due to excessive heat	5		3		105							0
19	belt/pulley: belt material durable for reasonable cycles of use	longitudinal rib cracking	complete loss of boost pressure	9	belt has been mistracked from groove, pulley groove tip has worn into tensile member	5		3		135							0
20	belt/pulley: belt material maintains dimensional properties for reasonable cycles of use	belt material elongates	decrease in boost pressure	7	incorrect selection of materials for strength property	5		3		105							0
21	belt/pulley: pulley shaft material durable for reasonable cycles of use	shaft material yields	complete loss of boost pressure	9	incorrect selection of materials for strength property	3		5		135							
22	belt/pulley: pulley shaft tolerance limits frictional losses	friction in shaft	decrease in boost pressure	7	incorrect selection of finish	3		7		147							0
23	belt/pulley: pulley shaft fits inside bearing with minimal frictional losses	friction in shaft	decrease in boost pressure	7	shaft installed at an angle with bearing	1		7		49							
24	belt/pulley: belt maintains alignment with pulley	groove jumping	complete loss of boost pressure	9	insufficient belt tension, incorrect pulley design, debris in pulley grooves, excessive belt speed, pulley misalignment, belt cord line is distorted	3		5		135							0
25	belt/pulley: belt material durable for reasonable cycles of use	belt snub break	complete loss of boost pressure	9	severe overtension, debris in between pulley and belt, belt turnover, severe misalignment, pulley or bearing failure	5		3		135							0

Row #	Item/Function	Potential Failure Mode	Potential Effect(s) of Failure	Severity (S)	Potential Cause(s)/ Mechanism(s) of Failure	Occurrence (O)	Current Design Controls/ Tests	Detection (D)	Recommended Actions	RPN	Recommended Actions	Responsibility & Target Completion Date	Action Results: Actions Taken				R
26	belt/pulley: belt quietly runs during life cycle	belt has objectionable squeak or causes	customer dissatisfaction	1	belt slippage, bearing noises, belt misalignment, system resonance induced vibration	3		1		3							0
	HYDRAULIC MOTOR																0
27	hydraulic motor: contains hydraulic fluid with no leakage	hydraulic fluid leaks	decrease in boost pressure	7	supplier design error, incorrect finish on mating parts	1		3		21							0
28	hydraulic motor: inlet flows hydraulic fluid with no turbulence or backflow	ID too small for flow path	decrease in boost pressure	7	design error, input exceeds compressor capacity	3		3		63							0
29	hydraulic motor: outlet flows hydraulic fluid with no turbulence or backflow	ID too small for flow path	decrease in boost pressure	7	design error, ouput exceeds compressor capacity	3		3		63							0
30	hydraulic motor: outlet transfers hydraulic fluid to reservoir	ID too small for flow path	decrease in boost pressure	7	design error, ouput exceeds reservoir capacity	3		3		63							0
31	hydraulic motor: outlet transfers hydraulic fluid to reservoir	coupling at outlet leaks hydraulic fluid	decrease in boost pressure	7	incorrect finish on mating parts	3		3	select a high viscosity hydraulic fluid	63							0
32	hydraulic motor: maintains high fluid pressures with varying loads/temperature	viscosity of fluid is low at high temperature, leading to pressure loss	decrease in boost pressure	7	incorrect selection of hydraulic fluid for operating temperatures	3		3		63							0
33	hydraulic motor: filter removes abrasive materials that degrade pump interior	filter failure leads to degradation of pump interior	decrease in boost pressure	7	filter failure	1		7		49							0
34	hydraulic motor: pump interior material withstands fluid stresses and	surface material flakes off	decrease in boost pressure	7	supplier design error, incorrect selection of materials for hardness property	1		7		49							0
	VALVE CONTROL																0
35	valve control: valve diameter varies flow	valve varying diameter becomes stuck	loss of control	9	supplier design error, incorrect dimensions	1		5		45							0
36	valve control: valve material durable for reasonable cycles	valve material yields	loss of control	9	supplier design error, incorrect selection of materials for valve	1		7		63							0
	USER INTERFACE																0
37	user interface: push-button control initiates on-demand boost	push-button becomes stuck	boost is longer than 6-7 seconds	9	supplier design error, incorrect dimensions	1		5		45							0
38	user interface: electronic wiring of push-button sends signal to hydraulic motor	electronic signal disturbed	complete loss of boost pressure	9	hydraulic fluid leaks and disturbs wiring	5		3		135							0

APPENDIX P: DFM&A

DESIGN FOR MANUFACTURABILITY / ASSEMBLY EVALUATION (DFM/A)

Product Name: ____________________

____________________ System

____________________ Subsystem Name: ____________________

____________________ Component

Prepared By Development Team: ____________________

____

DFM/A Origination Date: ____________________

Revisi

Row #	Criteria	Criteria Weight	Part / Assembly Description	Compressor	Hydraulic Motor	Belt/Pulley	User Interface	Valve Control		
			Multiplier:	1	1	2	1	1		
1	**Part Needed:** function can't be consolidated because 1. Part must be of a different material 2. Part must move relative to other parts 3. Part must be different to allow assembly or disassembly 4. Part is a purchased vendor catalog item	0 - Part must be separate - clearly meets one or more criteria 5 - Part seems to meet one criteria and, therefore, seems needed 10 - Part might be consolidated but significant trade-offs 15 - Part function could be performed in another way or part need can be eliminated		0	0	15	15	15		
2	**Part Fabrication or Subassembly:** part is easy to fabricate, adheres to DFM guidlines& has good yields. Geometric features are required & complexity justified by consolidation of multiple functions. Subassembly has minimum of parts and is easy to assemble.	0 - Part is easy to fabricate or assemble; no improvement opportunities 5 - Part fabrication or assembly can be improved 10 - Part difficult to fabricate or assemble; violates multiple guidlines		10	10	5	5	5		
3	**Part Handling:** part can be easily gripped or held from pick-up through insertion. Part is preserted or feed without interference with other parts, resting or targling. No effort to unpackage, remove protective material, or prepare. Part is not fragile or dangerous to handle.	0 - Part geometry makes it easy to grip and handle through insertion, no unpacking or preparation required 3 - May nesttangle, requires unpackaging or is difficult to handle 5 - Severe nesting, targling, or part interference, cannot be easily gripped, part is fragile, dangerous, or part requires special handling		5	3	0	0	0		
4	**Part Orientation:** orientation is unambiguous or has a high degree of symmetry that makes orientation easy. If part must be asymmetrical the features that define the asymmetry are obvious	0 - Part is oriented, unambiguous, or combined symmetry <180 deg 2 - Some ambiguity, symmetry <360 deg, or emphasized asymmetry 3 - Asymmetrical part, but asymmetrical features not as obvious 5 - Orientation not obvious, requires care to properly orient		5	0	5	0	0		
5	**Part Size and Weight:** size is neither too small or too large and heavy to handle manually. Does not require special tools (eg tweezer) nor another person to handle. Part can be packed-up with	0 - Part easy to pick-up using one hand without tools 3 - Part either small and difficult to grasp or large and requires 2 hands 5 - Part is very small and requires tool (eg tweezers) or very large and heavy and requires a tool, or 2 people to		5	3	3	3	3		
5	**Assembly Access:** top down assembly with a stable base component. No separate fixture required (self-fixturing). No reorientation of the assembly required to get access for assembly. No blind assembly; can be seen and guided by operator or machine.	0 - Top down assembly in the open; no reorientation or fixturing 2 - Side or bottom insertion or reorientation or fixturing required 4 - Blind assembly or access limitations 5 - Assembly requires significant efforts to reorient or fixture or blind assembly with severe access limitations		4	4	4	2	2		

DESIGN FOR MANUFACTURABILITY / ASSEMBLY EVALUATION (DFM/A)

Product Name: ____________________

____________________ System

____________________ Subsystem Name: ____________________

____________________ Component

Prepared By Development Team: ____________________

____

DFM/A Origination Date: ____________________

Revisio

Row #	Criteria	Criteria Weight	Part / Assembly Description	Compressor	Hydraulic Motor	Belt/Pulley	User Interface	Valve Control		
			Multiplier:	1	1	2	1	1		
6	**Part Insertion:** part is easily aligned and inserted with a simple, straight insertion direction, no insertion force and plenty of clearance to operator and tool. Part features facilitate alignment and insertion.	0 - Part is easily aligned and inserted with minimal force 2 - Part lacks alignment or insertion features or is flexible 4 - Requires feature/tool to align/insert, flexible, minimal clearance 5 - Difficult to insert, high insertion force, lacks clearance		4	4	5	4	4		
7	**Joining and Fastening:** requires minimal effort, integral attachment is the ideal with no separate fasteners or joining material. Avoid need for torquing or curing time. When fasteners required, use common fasteners.	0 - Part has integral attachment features or little fastening effort 3 - Part attached by separate fasteners or fastening material with moderate effort, fasteners common and conform to guidlines 5 - Significant joining/fastening effort, joining and fastening does not conform to guidlines and potential for fastening errors exist		3	3	3	3	3		
8	**Adjust and Finish:** assembly setup does not require any adjustment. Assembly step/part does not require cleaning or finishing. Assembly step/part does not require final cosmetic inspection.	0 - No adjustment, finish requirements, or special handling 3 - Part requires some adjustment, cleaning, finishing, or special handling 5 - Significant adjustment, calibration, cleaning, finishing.		0	0	3	5	5		
9	**Mistake-Proof:** product or process design features prevent part from being incorrectly assembled and avoid the need for subsequent inspection and checking.	0 - Part and assembly mistake-proofed, no check required 2 - Features to facilitate correct assembly, but error possible 3 - Features to detect errors in assembly.		3	3	3	3	3	**Grand Total**	
			Part/Operation Point Total	39	30	92	40	40	241	

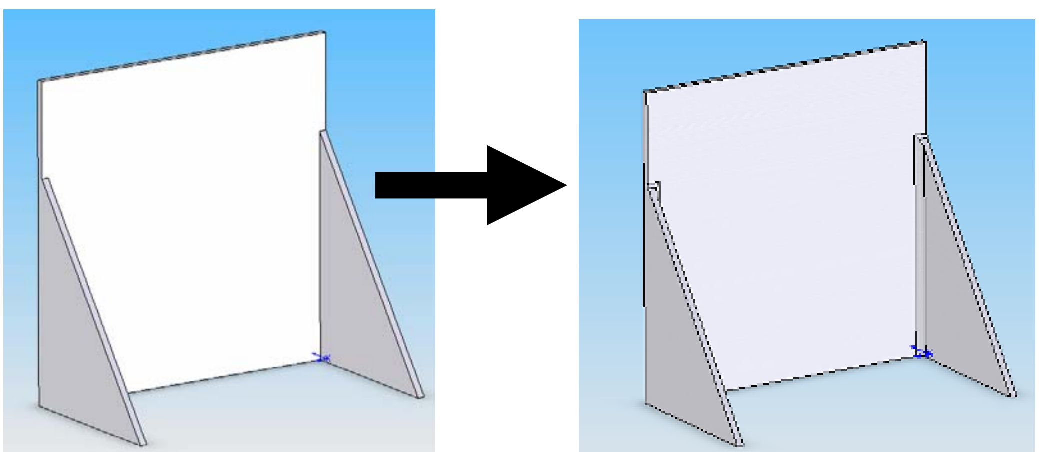
APPENDIX Q: Design Changes for Manufacturing and Assembly

In order for our design to truly be successful, we needed to go through a number of design iterations, some large and some small, in order to not only make our supercharger system perform well, but also to increase the ease of manufacturing and assembly. A number of these kinds of changes are detailed in this section.

Several steps were taken in an attempt to render a more efficient and effective assembly for our system design. These steps included revision methods included in the Design for Manufacturing and Assembly (DFMA) packet distributed by Professor Kazuhiro Saitou. A link to this document can be found in Appendix P.

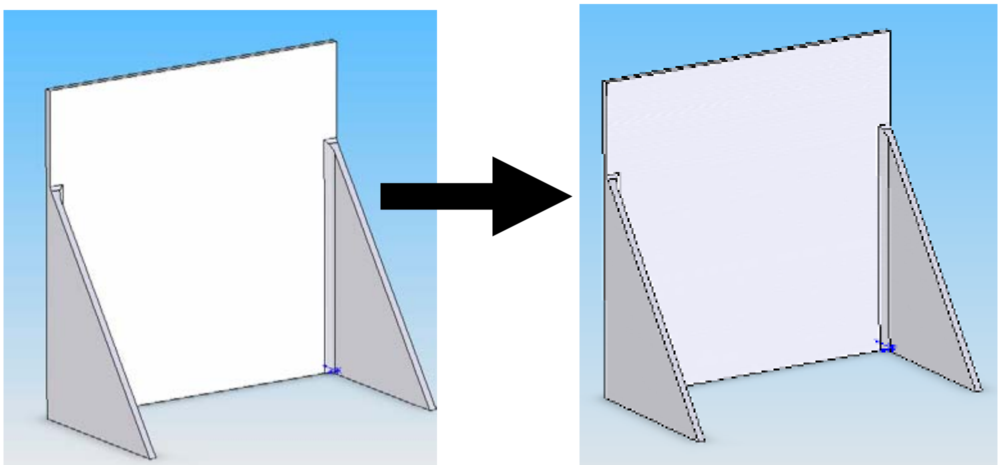
Below are some examples of where our group utilized DFMA technique to solve design problems within our system. The first few of these changes deal with the bracket which is used to support the sheave on the motor side of the assembly. Specifically, this change corresponds to DFMC-7 (Give R to Internal Corners), since the initial design was flawed. The bracket shown on the left in Figure 19 below has a sharp internal corner. To improve this, the bracket on the right has been given fillets remove the sharp corner created when two plates meet. In this case the two plates must be welded together to provide a secure base to mount the hydraulic motor.

Figure 19: Addition of fillets to bracket



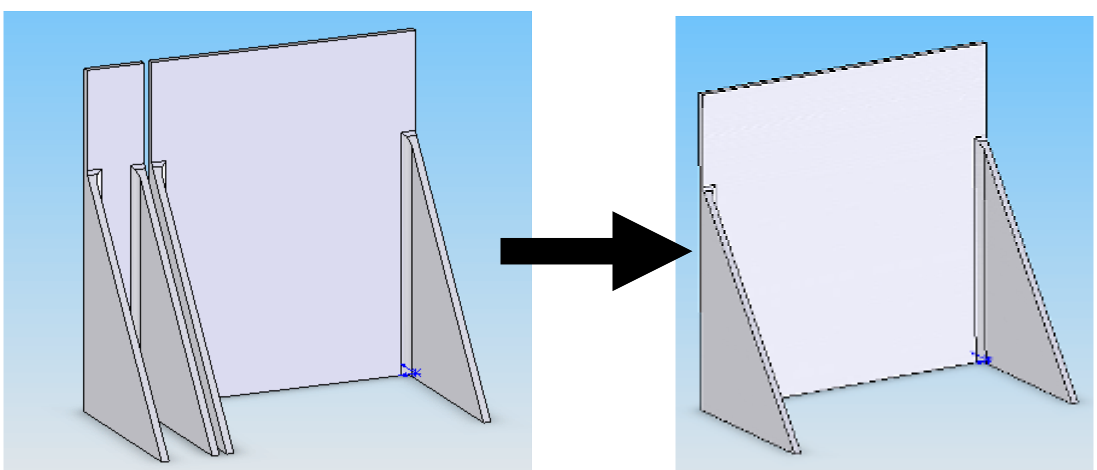
The second change we made deals with DFMC- (Assign Material for Ease of Manufacture), as Figure 20 below shows the same bracket with different material characteristics. The aluminum plate on the left is changed to the darker steel on the right. This added rigidity, strength and superior welding properties create a bracket with more robust design parameters.

Figure 20: Change of material from aluminum to steel



The third design modification we made to the design of this bracket is that our original design was more complex than it needed to be. In order to take into account DFAs-1 (Minimize Part count), we aimed to minimize complexity by combining a two bracket setup for the belt tensioner, where the two plates slide relative to each other, into to one bracket which includes an attached rotating belt tensioner system, as can be seen in the assembly drawing in Figure 4 earlier. While this might cause a slight cost increase, the modification is worthwhile due to its simplicity.

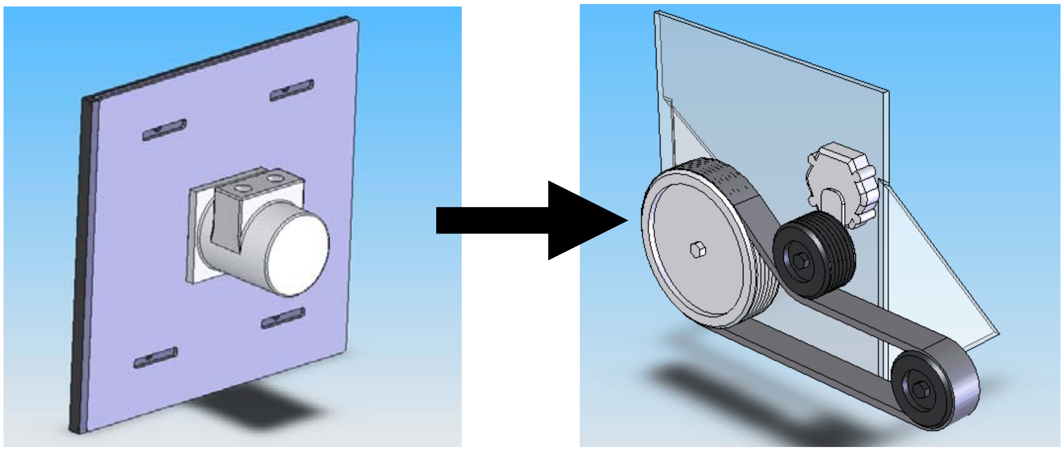
Figure 21: Subtraction of second bracket



The next set of components that required design iteration were all of the parts needed for the belt tensioner, as we moved from the drawing on the left in Figure 22 below to the one on the right. The modifications made here include DFAS-1 (Minimize Part Count), DFPI-2 (Eliminate Fasteners), as well as DFJ-1 (Add Alignment).

Several cases allowed for multiple DFMA-oriented changes. Features included in the belt tensioner on the left are fairly complex compared to the design on the right. The addition of a bracket with a stock fastener allowed us to minimize the number of parts and fasteners. Also, because the bracket holds the mounting points for two interacting components, we established a new means of alignment.

Figure 22: Redesign of belt tensioning system



APPENDIX R: Engineering Analysis Equations

Determination of all of the engineering specifications was completed in a number of steps. First we needed to do a thorough analysis of the customer requirements provided, specifically the pressure and flow targets. Using these targets as our design drivers, from this information we were able to begin the creation of a spreadsheet detailing all of our engineering specifications related to the performance of the system. First, we entered all of the fundamental data related to the fluid mechanics and thermodynamics involved with the operation of the supercharger. We assume that the air intake of the supercharger is at the ambient air pressure, 101 kPa. We also use quantities associated with standard air temperature and pressure. Specifically, we obtained values for the air's density (ρ), specific weight (γ), dynamic viscosity (μ), kinematic viscosity (ν), gas constant (R), and specific heat ratio (k). Here we will consider all of these as knowns, and all of these values may be seen in the section containing a more detailed development of the equations relating to engineering analysis, in Appendix R. We will be considering the case of the supercharger to be an adiabatic, isentropic process, which results in $PV^n = PV^k$, therefore n is equal to k, which has a value of 1.4, which may be utilized in the next steps.

The development of our exact specifications requires multiple steps in that we needed to know what might be possible in an ideal system first. Then from that point, using appropriate components and their associated performance characteristics, we needed to find out how a real system will perform, specifically we need to characterize the performance of our prototype system in order to make a valid comparison.

The first step is to analyze the performance of an ideal system. Ideal or not, the system we are creating must satisfy the engineering specifications we have set out previously in this report. Specifically, we need the supercharging unit to take the ambient air pressure, 101 kPa, and add to this the additional pressure we want to create at the output of the supercharger, 33 kPa, resulting in an absolute output pressure of 134 kPa. From this information we were able to move forward in our calculations, first by looking at the ratio of output to input pressure.

$$R_P = \left(\frac{P_{o,absolute}}{P_{i,absolute}} \right)_{SC} \approx 1.327 \qquad (1)$$

This equation then allowed for calculation of the ratios of the output to input temperature and volume based on the value of n=1.4, as discussed above.

$$R_V = R_P^{(1/n)} \approx 1.224 \qquad (2) \qquad\qquad R_T = R_V^{(n-1)} \approx 1.084 \qquad (3)$$

Knowing these ratios and knowing the values for the input density and temperature, we are then able to find these quantities for the output. However, these facts are not critical here so they are left to Appendix R. What is important is to take into account not only the pressure rise we need to create, but also the desired volume flow rate. We were provided with the figure for the input and output standard flow rate Q=113 standard cubic feet per minute (scfm). We assume that the inlet of the supercharger is at standard air temperature and pressure, so the inlet volume flow rate in actual cubic feet per minute (acfm) will be equal to the 113 scfm given. However, we know that the outlet will be pressurized and will be at a higher temperature. Thus the supercharger outlet and will therefore have a different value for its flow rate in acfm, which we calculated to be about 137.7 acfm based on Equation 2 above.

$$Q_o = (Q_i \times R_V)_{SC} \approx 137.7\ acfm \qquad (4)$$

For use in further calculations, both the input and output flow rates were converted to a number of different sets of units. The inlet and outlet air velocities, as well as the mass flow rate of air through the supercharger may also be calculated at this point by dividing the volume flow rate by the cross-sectional area of the inlet and outlet, but this information is not necessary for the evaluation of the idealized performance of the system. Even in the case of the real system, as long as the inlet and outlet have a sufficient area, there should be virtually no losses since there will be no clogging of the air as there would be in the event that the area was too small, as could be imagined in the limiting case that the outlet was infinitely small.

The next step in the necessary analysis is to find the adiabatic head H_{ad} produced by the supercharger in pressurizing the air, which requires calculation of the quantity ZRT as shown below in equation 5.

$$ZRT = P_{o,abs} \times v_o \approx 89477\ J/kg \qquad (5)$$

This quantity is needed to use equation 6 to find H_{ad}.

$$H_{ad} = ZRT \times \frac{k}{k-1} \times \left(R_p^{(k-1)/k} - 1\right) \approx 25785\ m^2/s^2 \qquad (6)$$

This value for the adiabatic head is needed to find the minimum amount of power that could possibly be provided in order to supply the required pressure and flow rate for the supercharger. This ideal minimum power needed can be calculated by equation 7 below.

$$PWR = \left(\frac{H_{ad} \times Q_i \times 144 \times P_{i,abs}}{\eta_{ad} \times T_{abs} \times R_{air} \times 33000}\right) \approx 2.260\ HP \qquad (7)$$

This can be used later in the evaluation of the efficiency of the prototype and final design in that the actual power required will be greater than this minimum power.
The first step in the development of our prototype's exact theoretical specifications was to choose an appropriate supercharger and obtain its performance curves. We accomplished this with the Eaton MP45 model supercharger, for which we had plots of its inlet flow rate vs. speed rpm, power hp vs. speed, and delta T vs. speed. Each of these plots contains a curve corresponding to a boost pressure of 5 psi, which is 106.6% of our boost pressure goal of 4.689 psi, so this should be a good estimation of what we need to create, while giving us a little room for improvement.

With this knowledge, we were able to move forward and use linear interpolation on the performance plots to find the desired characteristics. First we used the previously provided inlet volume flow rate of 113 scfm to find the corresponding value for the supercharger's speed, about 5320 RPM. Knowing this, we were able to use linear interpolation on the next curve to find the corresponding power needed to be provided to drive the supercharger at the desired speed and pressure, which yielded a value of 3.956 horsepower (HP). Knowing this power, we were able to begin to move towards the calculation of the requirements for our power transfer and power supply systems.

With values known for the rotational speed of the supercharger and the power needed to be supplied, this allows for calculation of the torque needed to drive it as we desire. We used a conversion based on equation 8 below to find a value of about 3.906 ft-lb.

$$T_{in,SC} = \left(\frac{P_{in}}{\omega}\right)_{SC} \approx 3.906\ ft - lb \qquad (8)$$

With the torque into and rotational speed of the supercharger known, all that remains to be done is to select an appropriate hydraulic motor and design the power transfer system based on its performance characteristics, specifically its maximum efficiency point. Based on our power requirement for the input to the supercharger, we used a belt efficiency of 92% to calculate the hydraulic motor power output requirement to be about $P_{out,\,HM} = 4.245$ HP. Based on this power output and the speed ranges we are working with, we selected a motor with a maximum efficiency of about 88%. This efficiency should be achieved in the middle of its RPM range, since efficiency will be lost at either extreme. The motor is rated for 5550 RPM, therefore we will want to operate at or near $\omega_{HM} = 2775$ RPM. This allows for a calculation of the drive ratio by equation 9 below, as well as calculation of the torque output from the hydraulic motor by equation 10 below.

$$R_{Drive} = \left(\frac{\omega_{SC}}{\omega_{HM}} \right) \approx 1.909 \qquad (9) \qquad\qquad \mathrm{T}_{out,\,HM} = \left(\frac{P_{out}}{\omega} \right)_{HM} \approx 8.001\ ft - lb \qquad (10)$$

Knowing the desired drive ratio, we must decide on dimensions for the power transfer system, which we chose to be a belt drive system due to its existence on the supercharger we selected. This supercharger already has a pulley attached to it with a 2.4-inch diameter. Therefore, multiplying by the drive ratio, we find that the pulley attached to the hydraulic motor needs to have a diameter of about 4.6 inches. Any variation will simply make a slight change in the motor's speed and torque output that we will need to be aware of during the operation of the supercharger.

Now that we have both the ideal system and actual prototype system characterized, we are able to analyze its performance in terms of efficiency. We find based on equation 11 below that the efficiency of the supercharger unit in our prototype will have an efficiency of about 57.1%.

$$\eta_{SC} = \left(\frac{PWR_{\min}}{PWR_{actual}} \right)_{SC} \times 100\% \approx 57.1\% \qquad (11)$$

To find the total system efficiency we also needed to know the efficiency of the hydraulic motor. We used performance curves from the Oilgear variable-displacement hydraulic motor to understand the behavior of the efficiency of this type of motor as the displacement changes to various fractions of the full available displacement. These curves, as can be seen above in Figure 5 in section VIII, show that as the displacement decreases, so does the efficiency, and also shows that the best efficiency is generally located near the middle of the range of speeds of the motor.

From this curve, and knowing the range of operating pressures we will be dealing with, we were able to calculate the average displacement we will use in the hydraulic motor, which led to calculation of the average hydraulic motor efficiency as 67%. This relatively low result is due to the fact that we will be using on the order of 15%-30% of the available displacement in the hydraulic motor from our prototype design. Then multiplying this motor efficiency by the belt efficiency of 92% and the supercharger efficiency found earlier as 57.1%, we reach an overall minimum system efficiency of about 33.6% for our prototype design.

This efficiency can be improved in the final design by the fact that we selected a Lysholm twin-screw supercharger which would operate around 71% efficiency based on the performance curves found in Appendix S. It would also be improved by using the Oilgear hydraulic motor since its smaller maximum displacement will allow for more efficient operation around 75%. Using the same belt efficiency as before, 92%, we arrive then at a total system efficiency of around 49%. Thus the prototype is not too significant a setback from the final design, since it creates a total efficiency decrease of about 18%, and it can certainly be used as a proof-of-concept.

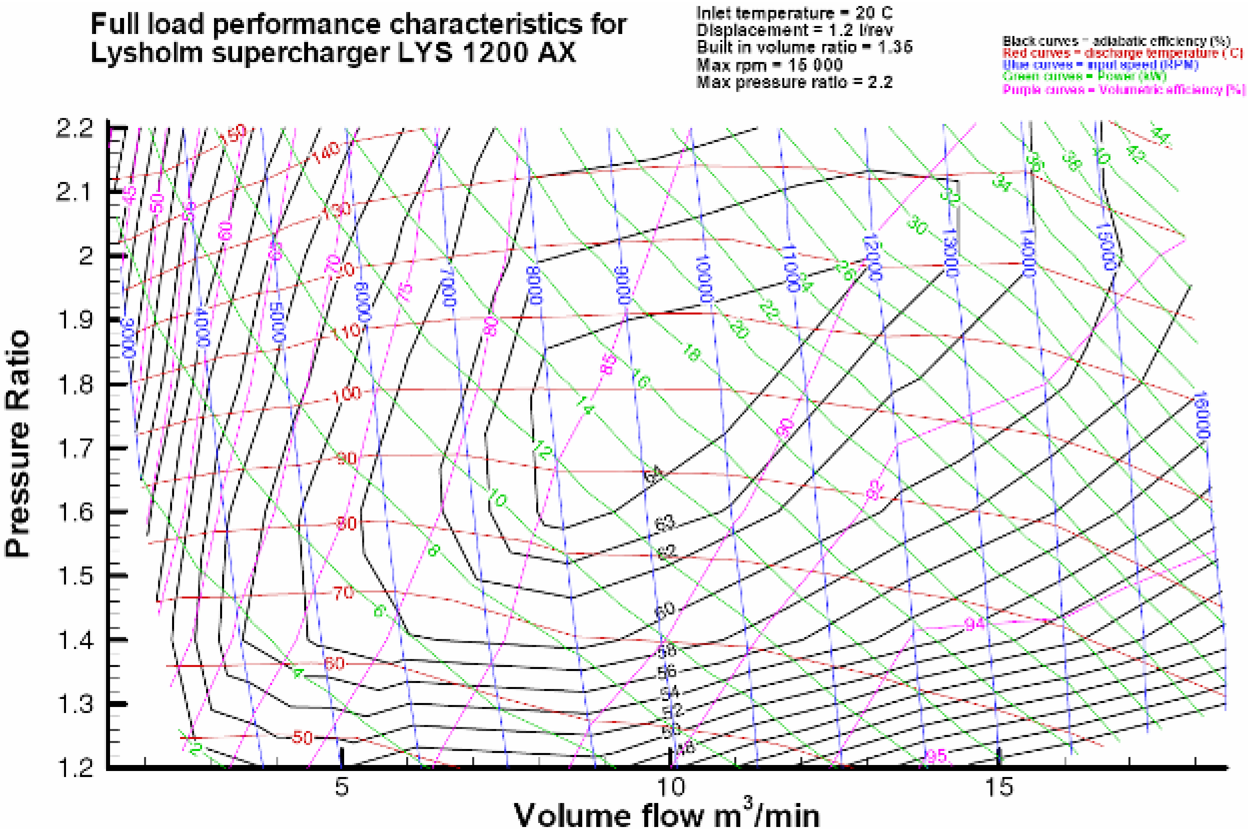
Other engineering specifications for both the final and prototype designs, such as material selection, have been completed by a simple logic process. In order to ensure that the hydraulic motor is constructed with sufficiently strong materials, we made sure that the component we selected for the prototype, the Bosch-Rexroth AA6VM variable-displacement 28 cc hydraulic motor, was rated for the full 5000 psi of hydraulic pressure that it may be supplied with. The same idea also applies to the selection of the Eaton MP45 supercharger unit, in that the pressures, flow rates, and RPM we will be using are within its specified limits, and in fact we will be operating this unit in the lower range of what it is capable of handling.

Some of the critical loaded parts which we will manufacture ourselves are the sheaves for the belt drive system and the mounts for this system. In order to minimize the risk of any pieces fracturing or fatiguing, we completed some elementary engineering analysis of the loads these pieces will see, and accounted for this by deciding on the appropriate part dimensions accordingly. In the case of the bending load put on the shaft connecting the hydraulic motor to the pulley, we need to account for the tension put in the belt. Here we need to worry not only about the strength of the shaft, but also the belt itself, as we do not want the belt to slip excessively, causing undue wear. To prevent this problem, we selected an appropriate belt based on its cross section and operational limit relative to our operating speeds.

For additional parts that we will be manufacturing ourselves, such as the mount for the supercharger system, belt tensioner, supercharger outlet manifold, and pulleys, the materials chosen are aluminum and steel. This is based on their tensile strength properties, low cost, low weight, and their ease of machining. Finite Element Analysis (FEA) was not necessary in verifying the suitability of using these materials, as the forces and moments applied are for the most part absorbed in the steel test stand.

The design is being modeled in component form with separate analysis of the supercharger system, the hydraulic motor system, and the energy transfer system. Then, these three sub-systems are unified to analyze the performance of the system as a whole. The level of analysis is appropriate in that all major design factors were taken into account, while auxiliary factors such as the rotational inertia induced losses in the energy transfer system were assumed negligible. These auxiliary factors did not affect the modeling of the system or the concept development process. We have confidence that our analysis is correct because we have taken into account all factors which we consider to greatly influence the performance of the supercharger. The analysis relates to our physical prototype in terms of the component selection process and energy transfer system characterization. The one place where we have made an engineering approximation is in the performance of the hydraulic motor. This is because we were unable to obtain the performance characterization curves for the motor we will be using in the prototype, only for the hydraulic motor in the final design. These engineering approximations are detailed in Appendix R. If we are able to obtain these performance curves, further analysis will be conducted to determine definitive characterization of the prototype system.

APPENDIX S: Lysholm Twin-Screw Supercharger Performance Curves

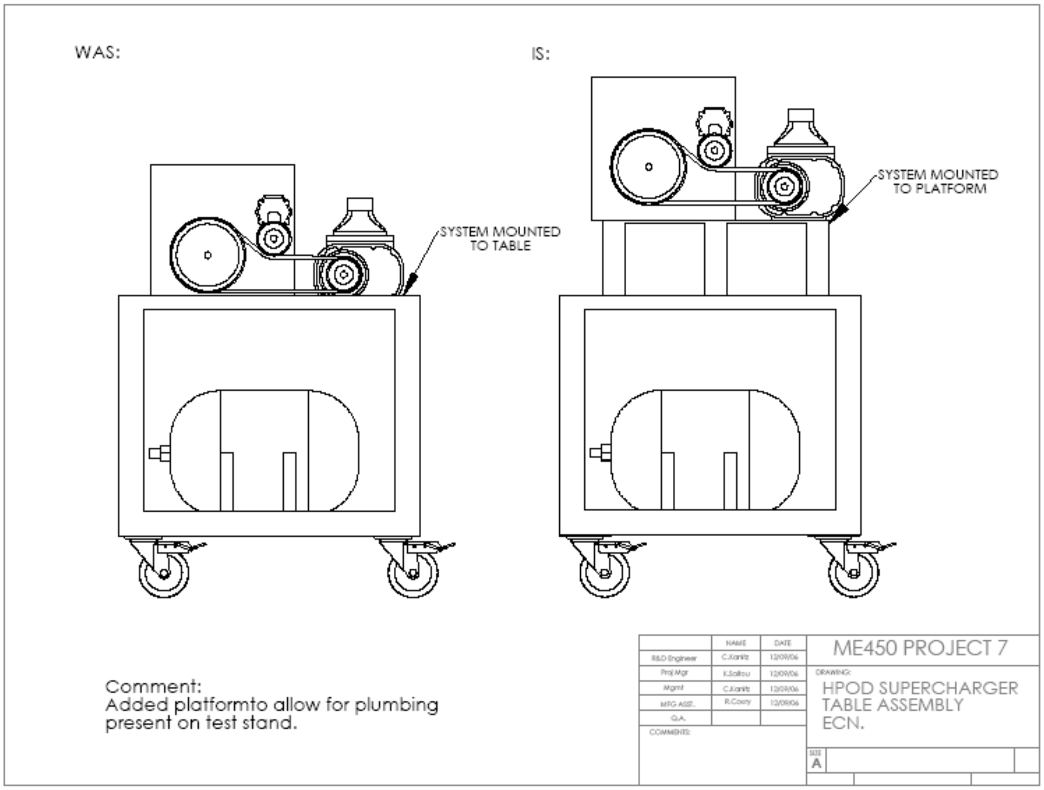


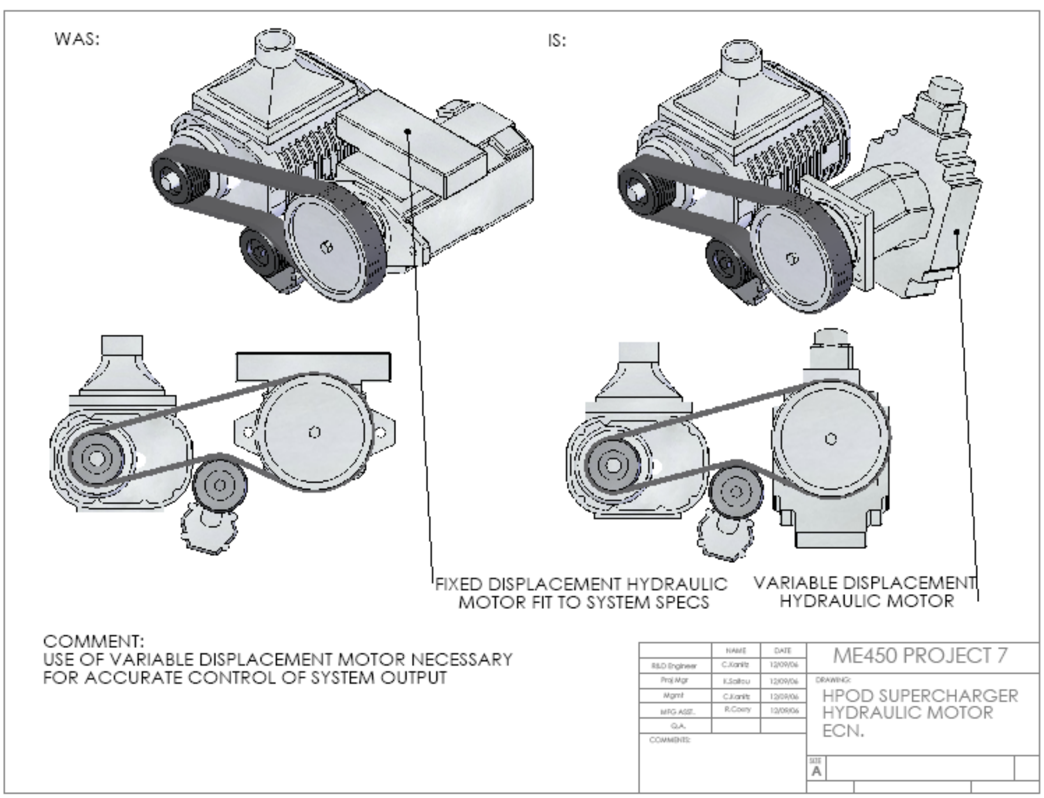
APPENDIX T: Prototype Bill of Materials

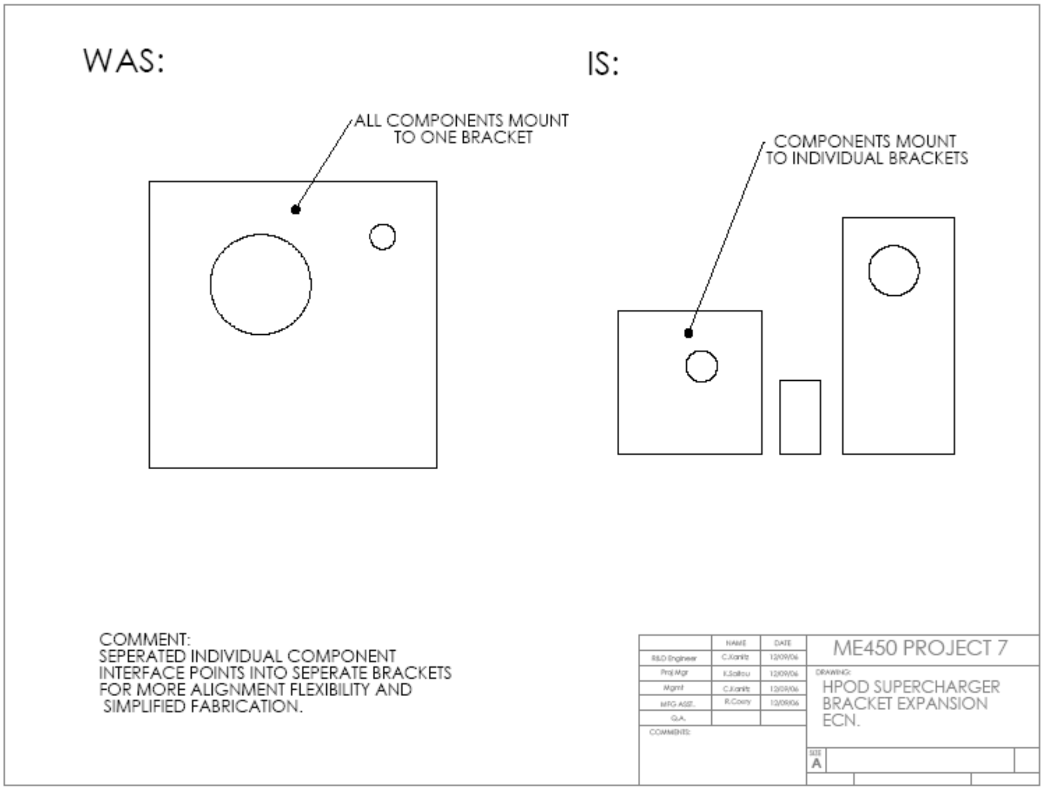
Quantity	Part Description	Purchased From	Part Number	Price (each)
1	Auto Belt Tensioner	AutoZone	#3696265	$36.99
1	“37 DuraLast Belt	AutoZone	#370K6-6PK940	$13.99
1	8” Diameter Aluminum Stock, “6 length	University of Michigan Shop	-	$0.00
1	“3 Diameter Aluminum Stock, 2” length	USEPA	-	$0.00
1	Aluminum Plates (13.5” x 24.0” x 0.5”) (10.0” x 7.0” x 0.50”)	Alro Metals Plus	EDP# AAA02500	$63.65 ($3.35/lb)
1	Aluminum 6061-T6, 1.5” Diameter Pipe, 12” length	Alro Metals Plus	EDP# 21203021	$18.75
1	Eaton MP45 Roots Type Supercharger	USEPA	#207050 DI04043	$0.00
1	Bosch-Rexroth AA6VM Variable Displacement Hydraulic Motor	USEPA	#9921564	$0.00
1	Steel Hydraulic Motor 2” Diameter Spline Coupling, 2” length	USEPA	-	$0.00
1	Spline Coupling Mount: Steel Plate (4.0” x “4.0 x “0.25)	University of Michigan Shop	-	$0.00
3	Supercharger, Motor, Belt Tensioner Mounts: Steel Plate (14.0” x “11.0 x 0.25”) (15.0” x 9.00” x 0.25”) (“8.00 x 4.00” x 0.25”)	University of Michigan Shop	-	$0.00
	1” Steel Brackets, 30” length	University of Michigan Shop		$0.00
18	5/8”-24 Screws	University of Michigan Shop	-	$0.00
11	Alloy 12.9 Screws	University of Michigan Shop	-	$0.00
10	Rockford Screws	University of Michigan Shop	-	$0.00
2	Rockford Nuts	Carpenter Bros. Hardware	-	$1.50 ($0.75 each)
1	3/8”-24 Screw	Carpenter Bros. Hardware	-	$1.35
4	1.75” Diameter Mounting Legs, “7 length each	USEPA	-	$0.00
1	Painter’s Tape	The Home Depot	-	$5.97
5	Spray Paint (Blue, Gold, Paint Primer, Steel Primer, Enamel)	The Home Depot	-	$18.79
1	Sandpaper (400 and 120 grit)	University of	-	$0.00

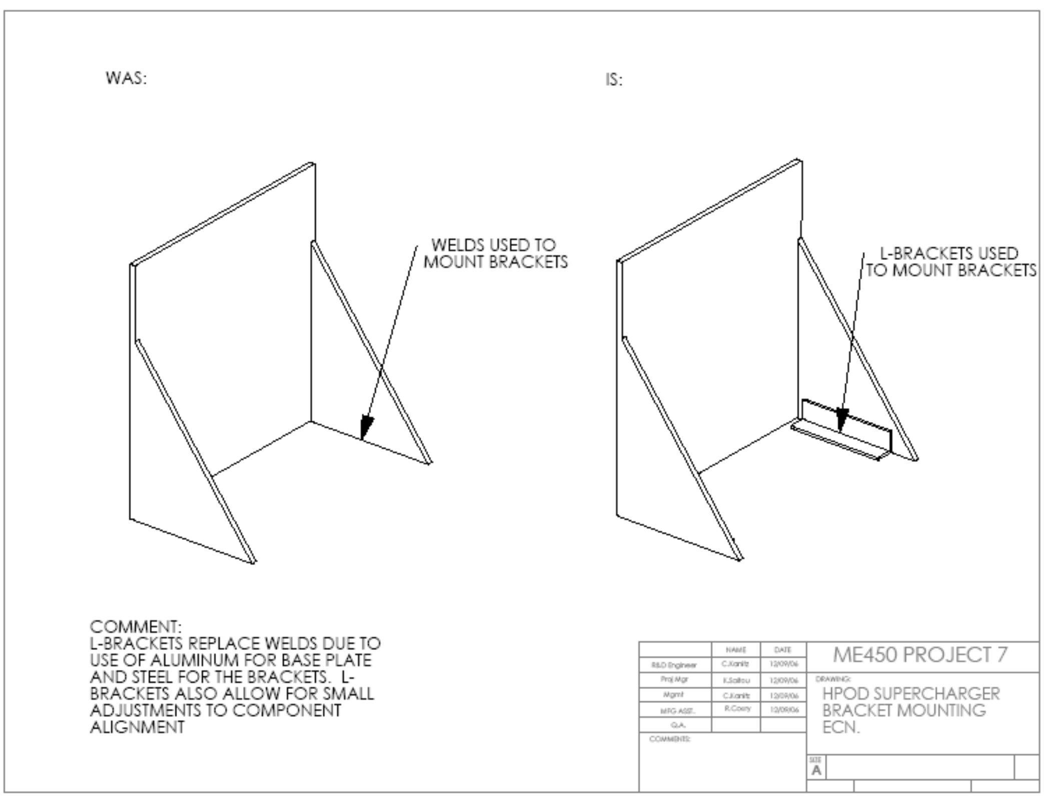
		Michigan Shop		
1	Stainless Steel Cleaner	The Home Depot	-	$4.97
1	Design Expo Poster Clips	Stapes	-	$2.98
1	Clear Scotch Tape	The Home Depot	-	$0.99
1	Acetone Cleaner	University of Michigan Shop	-	$0.00
			Subtotal:	169.93
			Taxes:	10.20
			Total:	**180.13**

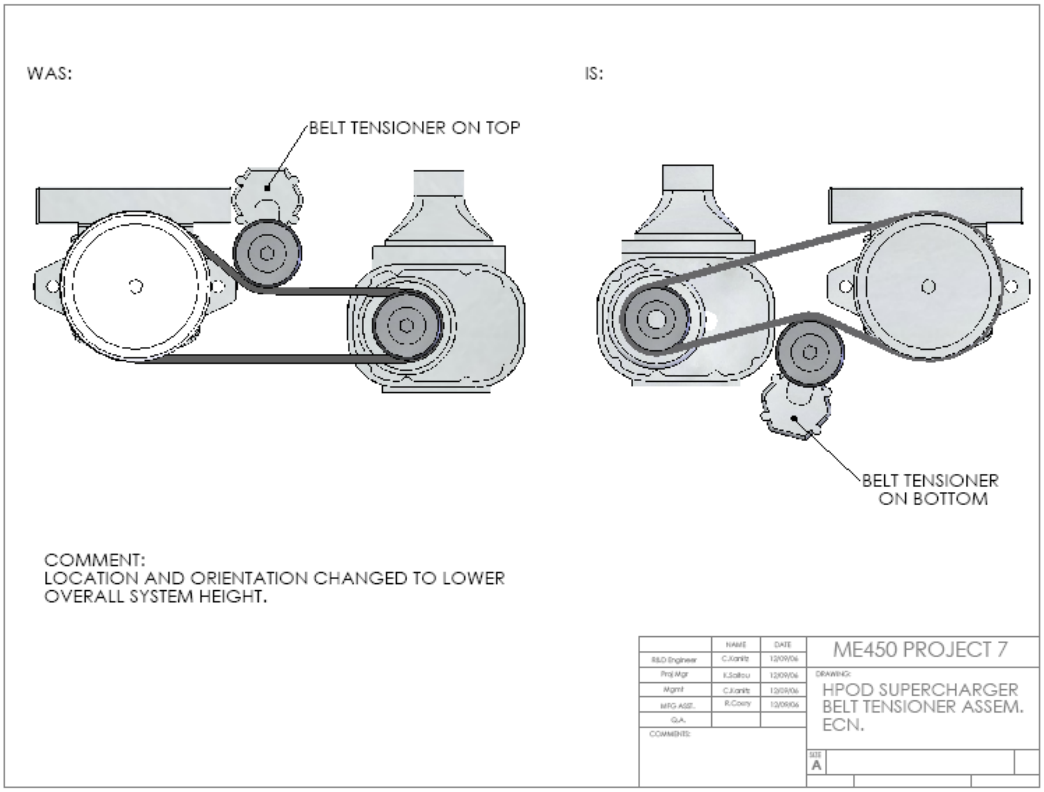
APPENDIX U: Engineering Changes Notice (ECN)

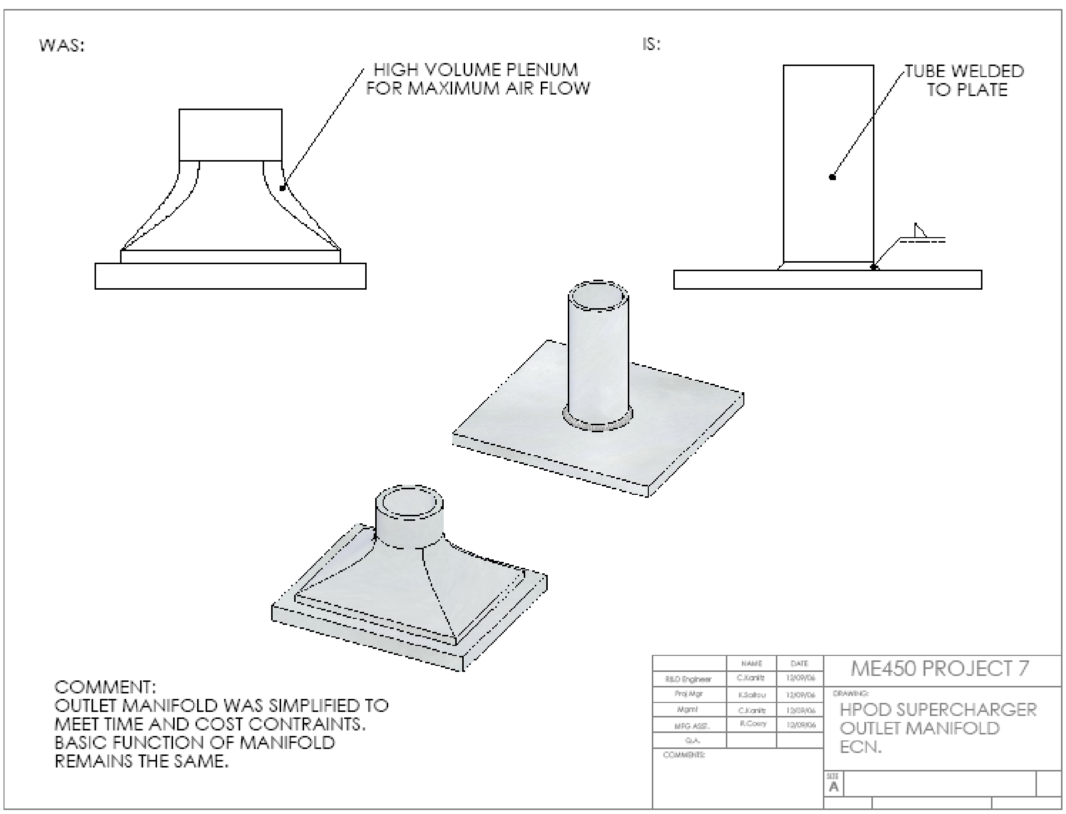


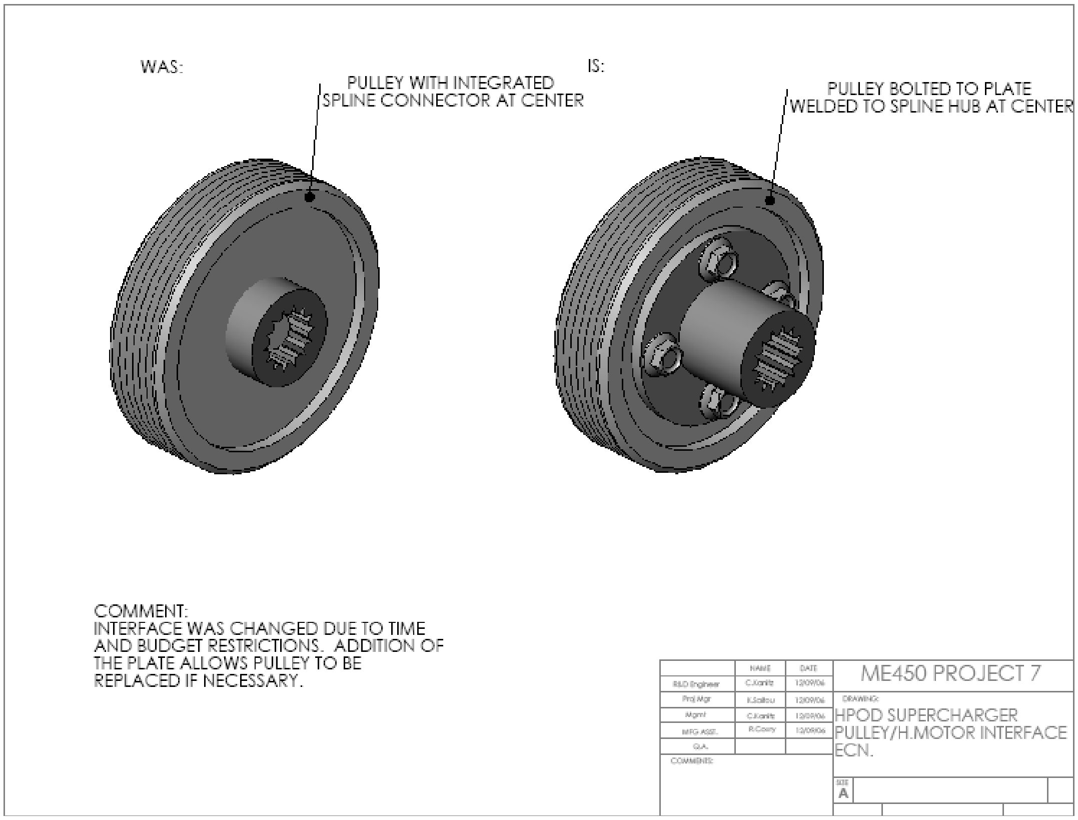


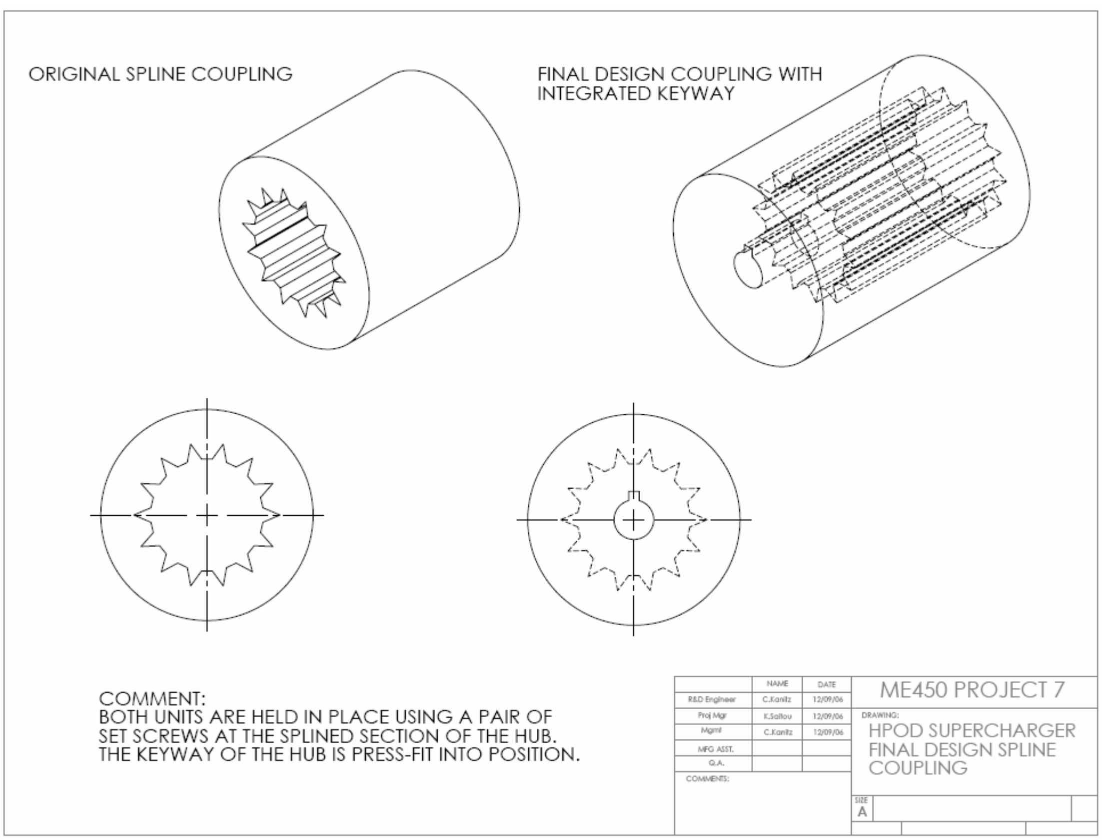

WAS:
IS:
ALL COMPONENTS MOUNT
TO ONE BRACKET
COMPONENTS MOUNT
TO INDIVIDUAL BRACKETS
COMMENT:
SEPERATED INDIVIDUAL COMPONENT
INTERFACE POINTS INTO SEPERATE BRACKETS
FOR MORE ALIGNMENT FLEXIBILITY AND
SIMPLIFIED FABRICATION.
ME450 PROJECT 7
DRAWING:
HPOD SUPERCHARGER
BRACKET EXPANSION
ECN.
SIZE
A


WAS:
IS:
WELDS USED TO
MOUNT BRACKETS
L-BRACKETS USED
TO MOUNT BRACKETS
COMMENT:
L-BRACKETS REPLACE WELDS DUE TO
USE OF ALUMINUM FOR BASE PLATE
AND STEEL FOR THE BRACKETS. L-
BRACKETS ALSO ALLOW FOR SMALL
ADJUSTMENTS TO COMPONENT
ALIGNMENT
ME450 PROJECT 7
DRAWING:
HPOD SUPERCHARGER
BRACKET MOUNTING
ECN.
SIZE
A


WAS:
IS:
BELT TENSIONER ON TOP
BELT TENSIONER
ON BOTTOM
COMMENT:
LOCATION AND ORIENTATION CHANGED TO LOWER
OVERALL SYSTEM HEIGHT.
NAME
DATE
R&D Engineer
12/09/06
Proj Mgr
12/09/06
Mgmt
12/09/06
MFG ASST.
12/09/06
Q.A.
COMMENTS:
ME450 PROJECT 7
DRAWING:
HPOD SUPERCHARGER
BELT TENSIONER ASSEM.
ECN.
SIZE
A


WAS:
IS:
HIGH VOLUME PLENUM
FOR MAXIMUM AIR FLOW
TUBE WELDED
TO PLATE
COMMENT:
OUTLET MANIFOLD WAS SIMPLIFIED TO
MEET TIME AND COST CONTRAINTS.
BASIC FUNCTION OF MANIFOLD
REMAINS THE SAME.
NAME
DATE
R&D Engineer
12/09/06
Proj Mgr
12/09/06
Mgmt
12/09/06
MFG ASST.
12/09/06
Q.A.
COMMENTS:
ME450 PROJECT 7
DRAWING:
HPOD SUPERCHARGER
OUTLET MANIFOLD
ECN.
SIZE
A


WAS:
PULLEY WITH INTEGRATED SPLINE CONNECTOR AT CENTER
IS:
PULLEY BOLTED TO PLATE WELDED TO SPLINE HUB AT CENTER
COMMENT:
INTERFACE WAS CHANGED DUE TO TIME AND BUDGET RESTRICTIONS. ADDITION OF THE PLATE ALLOWS PULLEY TO BE REPLACED IF NECESSARY.
NAME
DATE
R&D Engineer
C.Kanitz
12/09/06
Proj Mgr
K.Saitou
12/09/06
Mgmt
C.Kanitz
12/09/06
MFG ASST.
R.Coury
12/09/06
Q.A.
COMMENTS:
ME450 PROJECT 7
DRAWING:
HPOD SUPERCHARGER PULLEY/H.MOTOR INTERFACE ECN.
SIZE
A


ORIGINAL SPLINE COUPLING
FINAL DESIGN COUPLING WITH INTEGRATED KEYWAY
COMMENT:
BOTH UNITS ARE HELD IN PLACE USING A PAIR OF SET SCREWS AT THE SPLINED SECTION OF THE HUB. THE KEYWAY OF THE HUB IS PRESS-FIT INTO POSITION.
NAME
DATE
R&D Engineer
C.Kanitz
12/09/06
Proj Mgr
K.Saitou
12/09/06
Mgmt
C.Kanitz
12/09/06
MFG ASST.
Q.A.
COMMENTS:
ME450 PROJECT 7
DRAWING:
HPOD SUPERCHARGER FINAL DESIGN SPLINE COUPLING
SIZE
A

APPENDIX V: Engineering Specifications for Hydraulic Motor

Industrial Hydraulics | Electric Drives and Controls | Linear Motion and Assembly Technologies | Pneumatics | Service Automation | Mobile Hydraulics

Rexroth
Bosch Group

Axial Piston Variable Displacement Motor AA6VM (A6VM)

RA 91 604/02.04 1/64
replaces: 05.00

Open and closed circuits

Sizes 28 to 1000
Series 6

Sizes 28 to 200	Nominal pressure	5800 psi (400 bar)
	Maximum pressure	6500 psi (450 bar)
Sizes 250 to 1000	Nominal pressure	5100 psi (350 bar)
	Maximum pressure	5800 psi (400 bar)

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Features

- Variable displacement axial piston motor of bent axis design for hydrostatic transmissions in open and closed circuits
- For use in mobile and stationary applications
- The wide control range enables the variable displacement motor to satisfy the requirement for high rotational speed and high torque.
- The displacement is continuously variable from $V_{g\,max}$ to $V_{g\,min} = 0$.
- The output speed depends on the flow capacity of the pumps and the displacement of the motor.
- The torque increases with the pressure differential between the high and low pressure side and with increasing displacement.
- Wide control range with hydrostatic transmissions
- Wide selection of regulating and control devices
- Cost savings as no need for shiftable gearboxes and possibility to use smaller pumps
- Rugged, compact bearing system with long service life
- High power density
- Favorable start-up efficiency
- Low moment of inertia
- Large swivel range

Technical Data

Pressure fluid

Before starting project planning, please refer to our data sheets RA 90220 (mineral oil), RA 90221 (environmentally-friendly pressure fluids) and RA 90223 (HF pressure fluids) for detailed information regarding the choice of pressure fluids and conditions of use.

The AA6VM variable displacement motor is not suitable for use with HFA. If HFB, HFC and HFD or environmentally-friendly pressure fluids are being used, the constraints regarding technical data and seals mentioned in RA 90221 and RA 90223 must be observed.
If necessary, please contact us to discuss the type of pressure fluid you intend to use.

Viscosity range

We recommend that a viscosity (at operating temperature) for optimum efficiency and service life purposes of

ν_{opt} = optimum viscosity 80...170 SUS (16 to 36 mm²/s)

be chosen, taken the circulation temperature (closed circuit) and reservoir temperature (open circuit) into account.

Limits of viscosity range

The following values apply in extreme cases:

Sizes 28 to 200:

ν_{min} = 42 SUS (5 mm²/s)
short-term (t < 3 min) at max. permitted temperature of t_{max} = +240°F (+115°C).

ν_{max} = 7400 SUS (1600mm²/s)
short-term (t < 3 min) on cold start (p < 435 psi / 30 bar, n ≤ 1000 rpm, t_{min} = -40°F / -40°C).

Sizes 250 to 1000:

ν_{min} = 60 SUS (10 mm²/s)
short-term (t < 3 min) at max. permitted leakage-oil temperature of t_{max} = +195°F (+90°C).

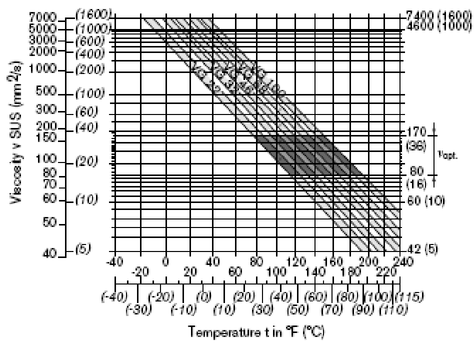
ν_{max} = 4600 SUS (1000 mm²/s)
short-term (t < 3 min) on cold start (p < 435 psi / 30 bar, n ≤ 1000 rpm, t_{min} = -13°F / -25°C).

Note that the maximum pressure fluid temperature must not be exceeded locally either (e.g. during storage).

Special measures are necessary at temperatures between -13°F and -40°F (-25°C and -40°C). Please contact us.

See RE 90300-03-B for detailed information about use at low temperatures.

Selection chart



Details regarding the choice of pressure fluid

The correct choice of pressure fluid requires knowledge of the operating temperature in relation to the ambient temperature: in a closed circuit the circulation temperature, in an open circuit the reservoir temperature.

The pressure fluid should be chosen so that the viscosity in the operating temperature range is within the optimum area (ν_{opt}) - the shaded area of the selection chart. We recommend that the higher viscosity class be selected in each case.

Example: At an operating temperature of 140°F (60°C), the viscosity classes VG 46 and VG 68 are within the optimum viscosity area (ν_{opt}, shaded field). In this case we would recommend
VG 68.

Please note: The leakage-oil temperature, which is affected by pressure and rotational speed, is always higher than the circulation temperature or reservoir temperature. At no point in the system must the temperature be higher than 240°F (115°C) for sizes 28 to 200 or 195°F (90°C) for sizes 250 to 1000.

If this cannot be achieved due to unusual operating parameters or high ambient temperatures, we recommend flushing of the case via port U or the use of a flushing and boost pressure valve (see page 55).

Technical Data

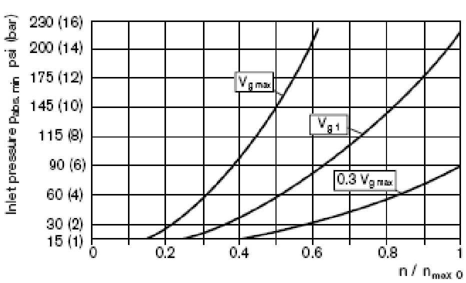
Table of values (theoretical values, ignoring η_{mh} and η_v; values rounded)

Size			28	55	80	107	140	160	200	250	355	500	1000
Displacement [1])	$V_{g\,max}$	in³	1.71	3.34	4.88	6.53	8.54	9.76	12.20	15.25	21.66	30.51	61.02
		cm³	28.1	54.8	80	107	140	160	200	250	355	500	1000
	$V_{g\,0}$	in³	0	0	0	0	0	0	0	0	0	0	0
		cm³	0	0	0	0	0	0	0	0	0	0	0
Rotational speed max. (while adhering to max. permitted flow)	n_{max} at $V_{g\,max}$	rpm	5550	4450	3900	3550	3250	3100	2900	2700	2240	2000	1600
	n_{max1} at $V_g < V_{g\,1}$	rpm	8750	7000	6150	5600	5150	4900	4600	3600	2950	2650	2100
	$V_{g,1}$	in³	1.10	2.14	3.11	4.15	5.37	6.16	7.69	11.47	16.48	23.00	46.5
		cm³	18	35	51	68	88	101	126	188	270	377	762
	$n_{max\,0}$ at $V_{g\,0}$	rpm	10450	8350	7350	6300	5750	5500	5100	3600	2950	2650	2100
Flow max.	$q_{V\,max}$	gpm	41	64	82	100	120	131	153	178	210	264	423
		L/min	156	244	312	380	455	496	580	675	795	1000	1600
Torque max.	T_{max} at $V_{g\,max}$ [2])	lb-ft	132	257	???	502	657	752	939	1026	1459	2054	4109
		Nm	179	349	509	681	891	1019	1273	1391	1978	2785	5571
Torsional rigidity		lb-ft/rad	266	516	848	1151	1545	1711	2146	2753	3756	6069	13832
		Nm/rad	360	700	1150	1560	2095	2320	2910	3733	5092	8228	18753
Mass moment of inertia J		lbs-ft²	0.033	0.100	0.190	0.301	0.491	0.600	0.838	1.448	2.420	4.224	13.05
around output shaft		kgm²	0.0014	0.0042	0.0080	0.0127	0.0207	0.0253	0.0353	0.061	0.102	0.178	0.550
Filling capacity		L	0.5	0.75	1.2	1.5	1.8	2.4	2.7	3.0	5.0	7.0	16.0
Mass (approx.)	m	lbs	35	57	75	104	132	141	176	198	375	463	948
		kg	16	26	34	47	60	64	80	90	170	210	430

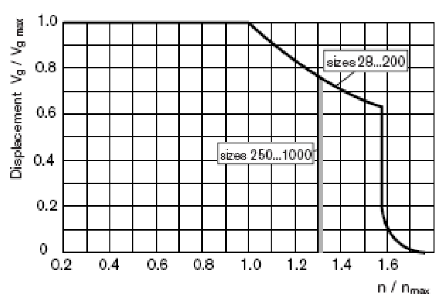
[1]) The minimum and maximum displacement are continuously variable, see model codes on page 2.
(default setting sizes 250 to 1000 unless specified in order: $V_{g\,min} = 0.2 \cdot V_{g\,max}$, $V_{g\,max} = V_{g\,max}$).

[2]) sizes 28 to 200: $\Delta p = 5800$ psi (400 bar); sizes 250 to 1000: $\Delta p = 5100$ psi (350 bar)

Minimum inlet pressure on service line port A(B)



Permitted displacement in relation to rotational speed



To prevent damage to the variable displacement motor, there has to be a minimum inlet pressure in the inlet area. The minimum inlet pressure depends on the speed and swivel angle (displacement) of the variable displacement motor.

Please contact us if these conditions cannot be satisfied.

Technical Data

Permissible transverse and axial forces on drive shaft

Size			28	55	80	107	140	160	200	250	355	500	1000
Transverse force, max [1]	$F_{q\,max}$	lbf	1280	2347		3434	4003	4568	5147	270[2]	337[2]	427[2]	584[2]
		N	5696	10440	13114	15278	17808	20320	22896	1200[2]	1500[2]	1900[2]	2600[2]
at distance of (from shaft collar)	a	in	0.49	0.59	0.69	0.79	0.89	0.89	0.98	1.61	2.07	2.07	2.66
		mm	12.5	15	17.5	20	22.5	22.5	25	41	52.5	52.5	67.5
Axial force, max. [3]	$-F_{ax\,max}$	lbf	71	112	160	202	231	252	281	270	337	427	584
		N	315	500	710	900	1030	1120	1250	1200	1500	1900	2600
	$+F_{ax\,max}$	lbf	71	112	160	202	231	252	281	899	1124	1405	2248
		N	315	500	710	900	1030	1120	1250	4000	5000	6250	10000
Permissible axial force/bar operating pressure	$\pm F_{ax\,perm.}$/bar	lbf/psi	0.07	0.12	0.15	0.18	0.21	0.23	0.26	[4]	[4]	[4]	[4]
	$\pm F_{ax\,perm.}$/psi	N/bar	4.6	7.5	9.6	11.3	13.3	15.1	17.0	[4]	[4]	[4]	[4]

1) During intermittent operation (sizes 28 to 200).
2) When stopped or when axial piston unit working in pressureless conditions. Higher forces are permitted when under pressure. Please contact us.
3) Max. permissible axial force when stopped or when axial piston unit working in pressureless conditions.
4) Please contact us.

When considering the permissible axial force, the force-transfer direction must be taken into account.

$-F_{ax\,max}$ = increase in service life of bearings

$+F_{ax\,max}$ = reduction in service life of bearings (avoid if at all possible)

Determining the size

Flow $q_V = \dfrac{V_g \cdot n}{231 \cdot \eta_v}$ gpm $\left(q_V = \dfrac{V_g \cdot n}{1000 \cdot \eta_v} \text{ L/min}\right)$

Output speed $n = \dfrac{q_V \cdot 231 \cdot \eta_v}{V_g}$ rpm $\left(n = \dfrac{q_V \cdot 1000 \cdot \eta_v}{V_g} \text{ rpm}\right)$

Output torque $T = \dfrac{V_g \cdot \Delta p \cdot \eta_{mh}}{24 \cdot \pi}$ lb-ft $\left(T = \dfrac{V_g \cdot \Delta p \cdot \eta_{mh}}{20 \cdot \pi} \text{ Nm}\right)$

Output power $P = \dfrac{2\pi \cdot T \cdot n}{33000} = \dfrac{q_V \cdot \Delta p \cdot \eta_t}{1714}$ HP

$$\left(P = \frac{2\pi \cdot T \cdot n}{60000} = \frac{q_V \cdot \Delta p \cdot \eta_t}{600} \text{ kW}\right)$$

V_g = Displacement per revolution in in^3 (cm^3)

Δp = Differential pressure in psi (bar)

η_v = Volumetric efficiency

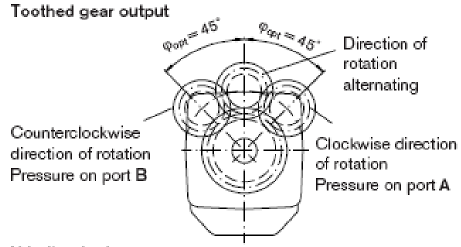
η_{mh} = Mechanical-hydraulic efficiency

η_t = Overall efficiency

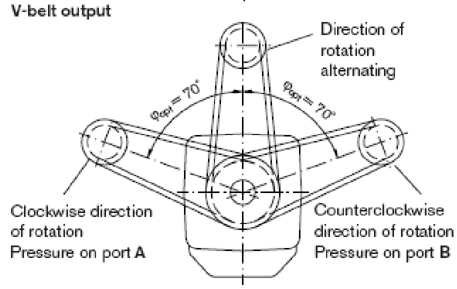
Effect of transverse force F_q on the service life of the bearings

By selecting a suitable force-transfer direction of F_q, the stress on the bearing caused by the internal transmission forces can be reduced, thus achieving the optimum service life for the bearing. Recommended position of mating gear depending on direction of rotation. Examples:

Toothed gear output

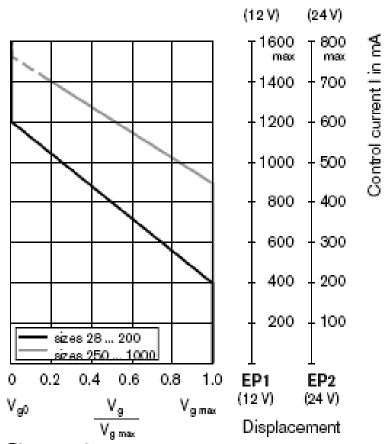


V-belt output



EP - Electrical Control With Proportional Solenoid

Electrical control using a proportional solenoid (sizes 28 to 200) or proportional valve (sizes 250 to 1000) permits continuous control of the displacement according to an electrical signal. The control is proportional to the applied electrical control current.
In the case of sizes 250 to 1000, an external pressure of p_{min} = 435 psi (30 bar) is necessary for the control oil supply to port P (p_{max} = 1450 psi (100 bar)).
Normal version:
– start of control at $V_{g\,max}$ (max. torque, min. speed)
– end of control at $V_{g\,min}$ (min. torque, max. permitted speed)



Please note:
- The required control oil is taken from the high pressure, so a Δp of at least 218 psi (15 bar) on the supply pressure is needed. If the Δp on the supply pressure is < 218 psi (15 bar) (when idle), an auxiliary pressure of at least 218 psi (15 bar) above the supply pressure must be applied on port G via an external check valve (valid for size 28...200, for size 250...1000 see page 11).
- The start of control and the EP characteristic are influenced by the pressure in the case. A rise in pressure in the case causes an increase in the start of control and a corresponding parallel movement of the performance curve (sizes 250 to 1000, see page 5).

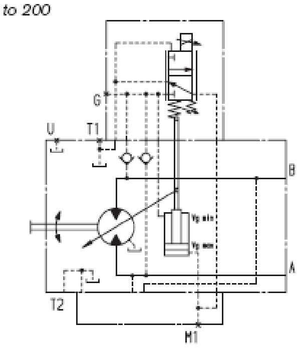
Technical data, solenoid in EP1, EP2	EP1	EP2
Voltage	12 V (±20 %)	24 V (±20 %)
Control current sizes 28 to 200		
Control starts at $V_{g\,max}$	400 mA	200 mA
Control ends at $V_{g\,min}$	1200 mA	600 mA
Control current sizes 250 to 1000		
Control starts at $V_{g\,max}$	900 mA	450 mA
Control ends at $V_{g\,min}$	1400 mA	700 mA
Maximum current	1,54 A	0,77 A
Nominal resistance (at 20°C)	5,5 Ω	22,7 Ω
Dither frequency	100 Hz	100 Hz
Operating time	100 %	100 %
Degree of protection	see connector design, page 60	

The rate of control or limiting of the displacement (limiting the swiveling range) can be achieved electrically using the following control units:

- **RC** control unit (see RE 95200)
- **PV** proportional amplifier (see RE 95023)
- **VT 2000** electrical amplifier, series 5X (see RE 29904) (for industrial application)

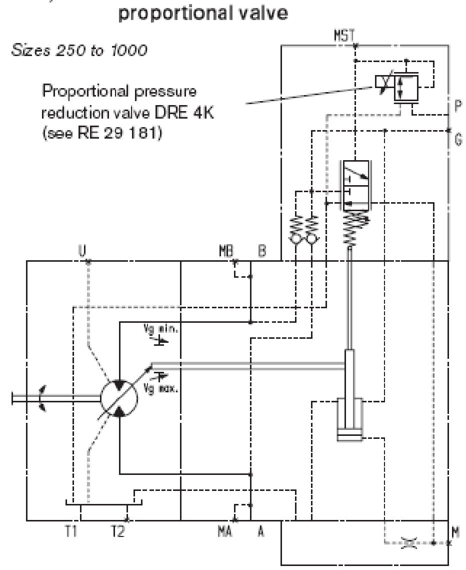
EP1, EP2: Electrical control with proportional solenoid

Sizes 28 to 200



EP1, EP2: Electrical control with proportional valve

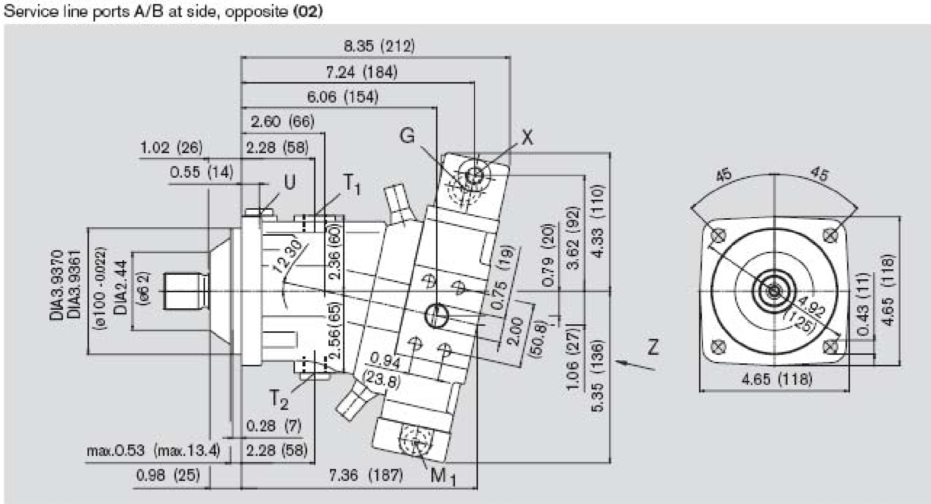
Sizes 250 to 1000



Unit Dimensions, Size 28 (ISO Version)

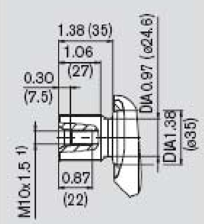
Before finalising your design, please request a certified drawing.
Dimensions in inches and (millimeters).

Hydraulic control, pilot pressure dependent HD1, HD2
Hydraulic two-point control HZ1
Service line ports A/B at side, opposite **(02)**

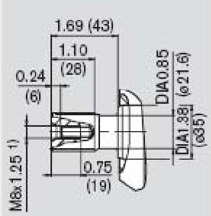


Shaft ends

A Splined shaft
DIN 5480
W30x2x30x14x9g

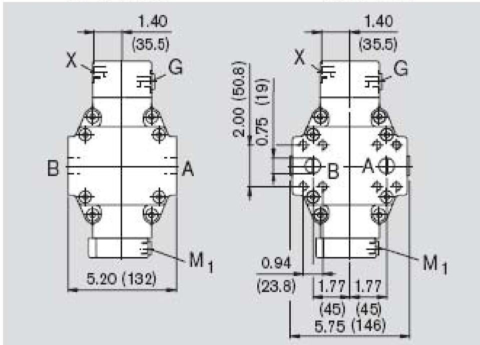


Z Splined shaft
DIN 5480
W25x1.25x30x18x9g



View Z

Service line ports A/B at side, opposite **(02)** | Service line ports A/B at rear **(01)**



1) DIN 332 center hole

Ports

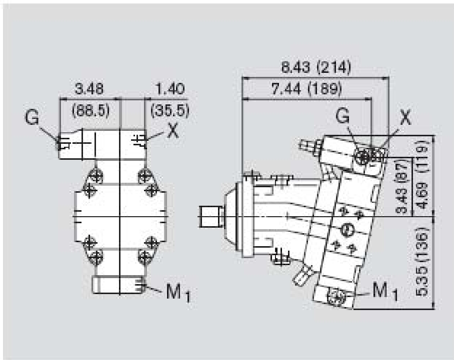
				Tightening torque, max. 3)
A, B	Service line ports (high pressure series)	SAE J518,	3/4 in	
	Threaded fitting A/B	DIN 13	M10x1.5; 0.67 (17) deep	see safety instructions
T_1	Leakage-oil port	DIN 3852	M18x1.5; 0.47 (12) deep	100 lb-ft (140 Nm)
T_2	Leakage fluid/oil drain 2)	DIN 3852	M18x1.5; 0.47 (12) deep	100 lb-ft (140 Nm)
X, X_1, X_3	Pilot pressure port	DIN 3852	M14x1.5; 0.47 (12) deep	60 lb-ft (80 Nm)
G	Port for synchronous control of several units and for remote charge pressure 2)	DIN 3852	M14x1.5; 0.47 (12) deep	60 lb-ft (80 Nm)
G_2	Port for 2nd pressure setting 2)	DIN 3852	M14x1.5; 0.47 (12) deep	60 lb-ft (80 Nm)
U	Flow port 2)	DIN 3852	M16x1.5; 0.47 (12) deep	70 lb-ft (100 Nm)
M_1	Measuring port for charge pressure 2)	DIN 3852	M14x1.5; 0.47 (12) deep	60 lb-ft (80 Nm)

2) plugged 3) note safety instructions, page 62

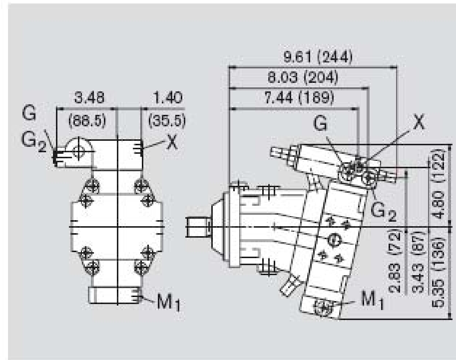
Unit Dimensions, Size 28 (ISO Version)

Before finalising your design, please request a certified drawing.
Dimensions in inches and (millimeters).

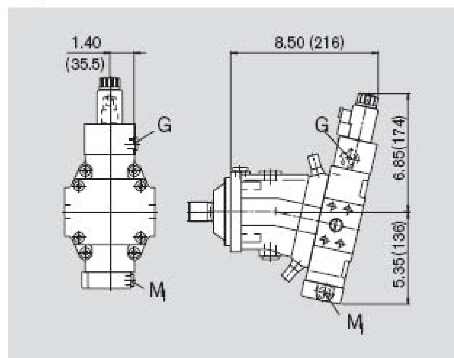
Hydraulic control, pilot pressure dependent, with pressure control, direct HD.D



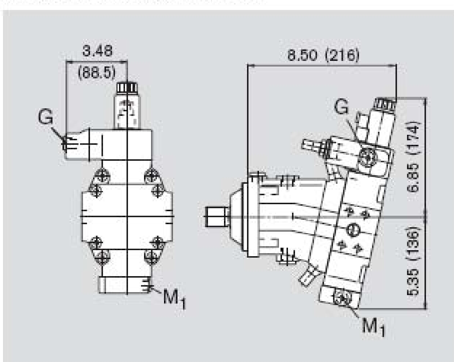
Hydraulic control, pilot pressure dependent, with pressure control, direct and 2nd pressure setting HD.E



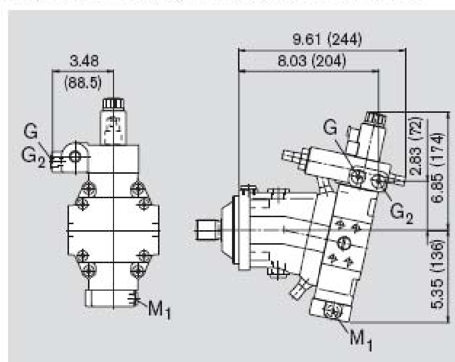
Electrical control with proportional solenoid EP1, EP2



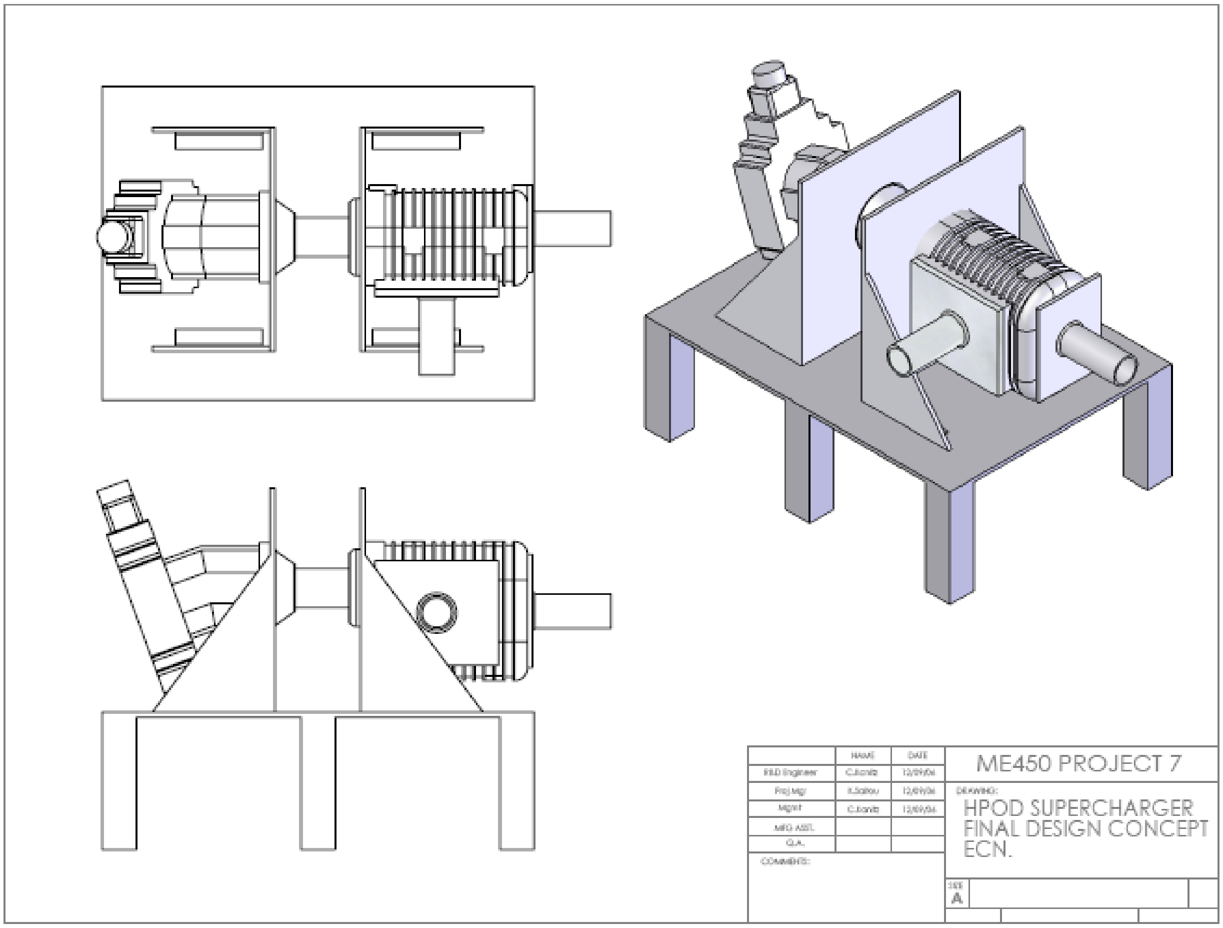
Electrical control (proportional solenoid) with pressure control, direct EP.D



Electrical control (proportional solenoid) with pressure control, direct and 2nd pressure setting EP.E



APPENDIX W: Prototype Final Design Layout



APPENDIX X: Supercharger Simulation Matlab Code

```
% Supercharger simulation
% ME 450 Project 7, Fall 2006
clear all; close all;
HPlow=200; % psi of Hydraulic Low-Pressure accumulator
HPhighmin=2000; % minimum psi of Hydraulic High-Pressure reservoir
HPhighmax=5000; % maximum psi of Hydraulic High-Pressure reservoir
deltaHPmax=HPhighmax-HPlow;
deltaHPmin=HPhighmin-HPlow;
HPdiff=10; % maximum step size of Hydraulic Pressure
%effM=1;
effMmax=1; % max efficiency of Hydraulic Motor
effBelt=0.92; % Belt Efficiency
effSC=0.591132058; % Supercharger Efficiency
ETorqMmin=7.864038602; % lb-ft Motor Torque at full boost
rpmM=2775.652174; % rpm rotational speed of motor at full boost
omegaM=rpmM*2*pi/60; % rad/s
omegaSCmax=5320*2*pi/60; % rad/s rotational speed of supercharger at full boost
PowerM=4.15601023; % Motor HP at full boost
PowerSC=PowerM*effBelt; % Supercharger HP at full boost
MPowerSCmin=2851.205385; % Power delivered to supercharger at full boost, in Watts
TorqSCmin=5.117858243; % N-m Torque on supercharger at full boost

% Calculate rotational inertia of system parts
IpulleySC=.5*2700*pi*1.2^4*1*0.0254^5; % kg-m^2
IpulleyHM=.5*2700*pi*2.3^4*1*0.0254^5; % kg-m^2
Iplate=.5*7840*pi*1.95^4*.25*0.0254^5; % kg-m^2
Iscrews=2*.5*7840*pi*1.25^4*6*0.0254^5; % kg-m^2
Ishafts=2*.5*7840*pi*.425^4*3*0.0254^5; % kg-m^2
Icoupling=.5*7840*pi*1^4*2*0.0254^5; % kg-m^2
Itensioner=930*pi*1^4*1*0.0254^5; % kg-m^2
I=IpulleySC+IpulleyHM+Iplate+Iscrews+Ishafts+Icoupling+Itensioner; % kg-m^2

% Set up maximums and timesteps to be used in loop
Dispmax=1.71; % in^3 maximum hydraulic motor displacement
Maxboost=5; % psi maximum boost
Qairmax=113; % cfm of air flow at full boost
deltat=0.002; % timestep for hydraulic motor response
t=[0:deltat:7]; %sec
n=length(t);
% Set up initial conditions
HPhigh(1)=HPhighmax;
deltaHP(1)=HPhigh(1)-HPlow;
Disp(1)=Dispmax; % in^3
%effM(1)=effMmax*(Disp(1)/Dispmax*100)^(1/2)*10;
```

```
effM(1)=effMmax*(1/100)*(Disp(1)/Dispmax*100)^0.2*39.81071705534973;
omegaSC(1)=5;
omegaHM(1)=omegaSC(1)*2.4/4.6;
SCboost(1)=5*omegaSC(1)/omegaSCmax; % psi
offset=0.22; % created for downward bias in hydraulic pressure

% Start loop at second timestep
for i=2:n
   % Find new Pressure in Hydraulic reservoir
   num=rand(2);
   if num(1,1)<=(num(1,2)+offset)
      HPhigh(i)=HPhigh(i-1)-HPdiff*num(1,1);
   else
      HPhigh(i)=HPhigh(i-1)+HPdiff*num(1,1);
   end
   % check that pressure does not exceed bounds
   % and if it does, find a new pressure
   while HPhigh(i)<HPhighmin | HPhigh(i)>HPhighmax
      num=rand(2);
      if num(1,1)<=(num(1,2)-offset)
         HPhigh(i)=HPhigh(i-1)-HPdiff*num(1,1);
      else
         HPhigh(i)=HPhigh(i-1)+HPdiff*num(1,1);
      end
   end
   deltaHP(i)=HPhigh(i)-HPlow; % difference in hydraulic pressure across the motor
   % find power going into supercharger based on displacement
   if SCboost(i-1) < (Maxboost-0.1)
      Disp(i)=Dispmax;
      ETorqM(i)=Disp(i)*(deltaHP(i)*effM(i-1))/(24*pi); %lb-ft
      MTorqM(i)=ETorqM(i)/0.73756215; % N-m
      MPowerM(i)=MTorqM(i)*omegaHM(i-1);
      MPowerSC(i)=MPowerM(i)*effBelt;
   else
      Disp(i)=24*pi*ETorqMmin/(deltaHP(i)*effM(i-1)); % in^3
      MPowerSC(i)=MPowerSCmin;
   End

   % Find boost and air flow rate
   %effM(i)=effMmax*(1/100)*(Disp(i)/Dispmax*100)^(1/2)*10;
   effM(i)=effMmax*(1/100)*(Disp(i)/Dispmax*100)^0.2*39.81071705534973;
   effSYS(i)=effM(i)*effBelt*effSC*100; % (%)
   TorqSC(i)=MPowerSC(i)/omegaSC(i-1); %N-m
   alphaSC(i)=(TorqSC(i)-TorqSCmin)/I;
   deltaomegaSC(i)=alphaSC(i)*deltat;
   omegaSC(i)=omegaSC(i-1)+deltaomegaSC(i);
```

```
   Qair(i)=Qairmax*omegaSC(i)/omegaSCmax;
   omegaHM(i)=omegaSC(i)*2.4/4.6;
   SCboost(i)=Maxboost*omegaSC(i)/omegaSCmax; % psi
end
% calculated quantities to plot
rpmSC=omegaSC*60/(2*pi);
EPowerSC=MPowerSC/745.69987;
kPaSCboost=SCboost*6.8947572931684;
for i=1:1001
   Displacement(i)=Dispmax*(i-1)/1000;
MotorEfficiency(i)=effMmax*(Displacement(i)/Dispmax*100)^0.2*39.81071705534973;
End

% make plots
plot(Displacement,MotorEfficiency,'k-');
xlabel('Hydraulic Motor Displacement (in^3)');
ylabel('Hydraulic Motor Efficiency (%)');
figure;
plot(t,deltaHP,'k-');
xlabel('time t (sec)');
ylabel('\Delta P Hydraulic');
figure;
plot(t,Disp,'k-');
xlabel('time t (sec)');
ylabel('Hydraulic Motor Displacement (in^3)');
figure;
plot(t,effM*100,'k-');
xlabel('time t (sec)');
ylabel('Hydraulic Motor Efficiency (%)');
figure;
plot(t,effSYS,'k-');
xlabel('time t (sec)');
ylabel('System Efficiency (%)');
figure;
plot(t,rpmSC,'k-');
xlabel('time t (sec)');
ylabel('Supercharger RPM');
figure;
plot(t,EPowerSC,'k-');
xlabel('time t (sec)');
ylabel('Supercharger Power in (HP)');
figure;
plotyy(t(1:250),kPaSCboost(1:250),t(1:250),Qair(1:250));
xlabel('time t (sec)');
ylabel('Supercharger boost (kPa)');
```

APPENDIX Y: Supercharger Simulation Plots

Figure 23: Supercharger Boost and Air Flow versus Time

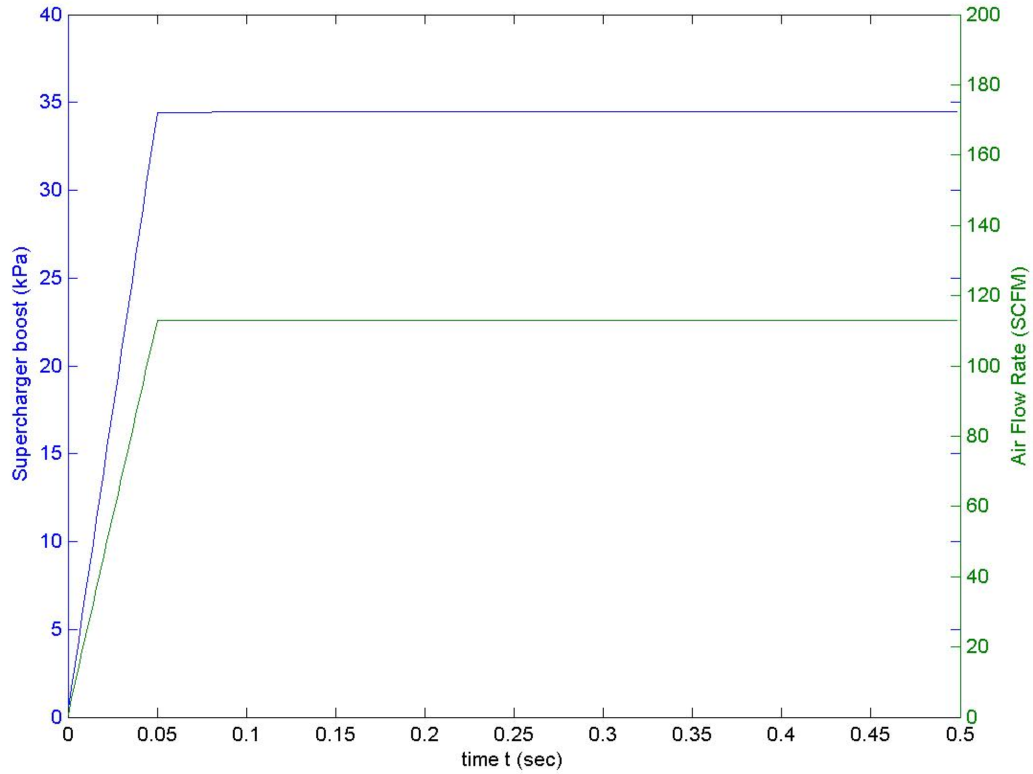


Figure 24: Supercharger Power Input versus Time

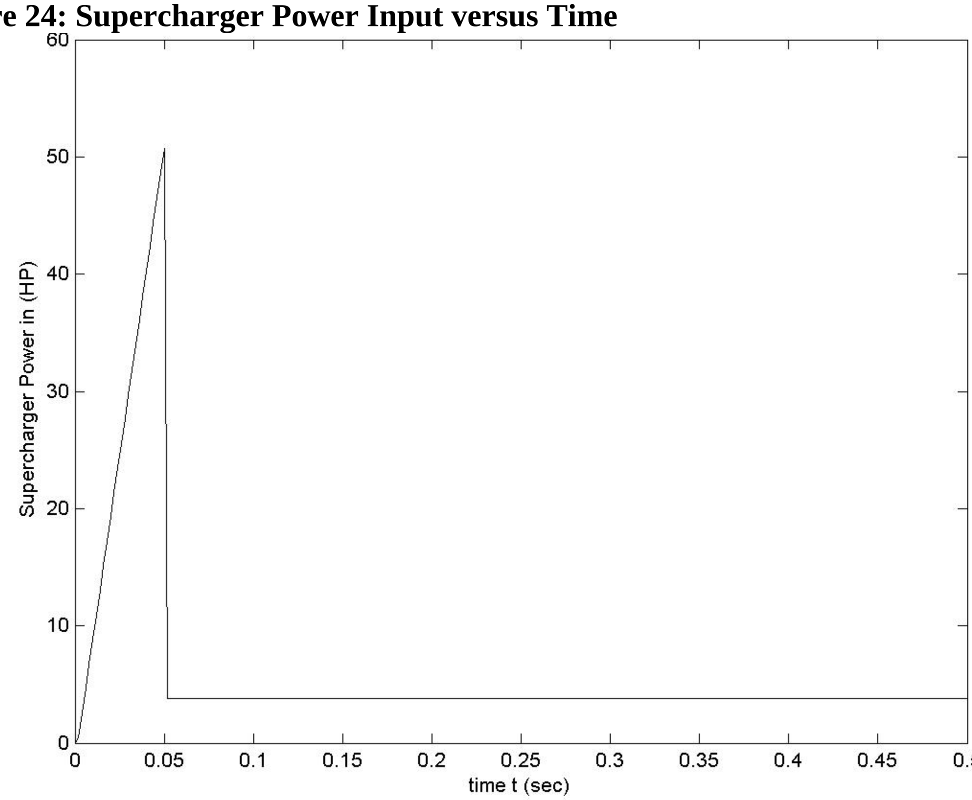


Figure 25: System Efficiency versus Time

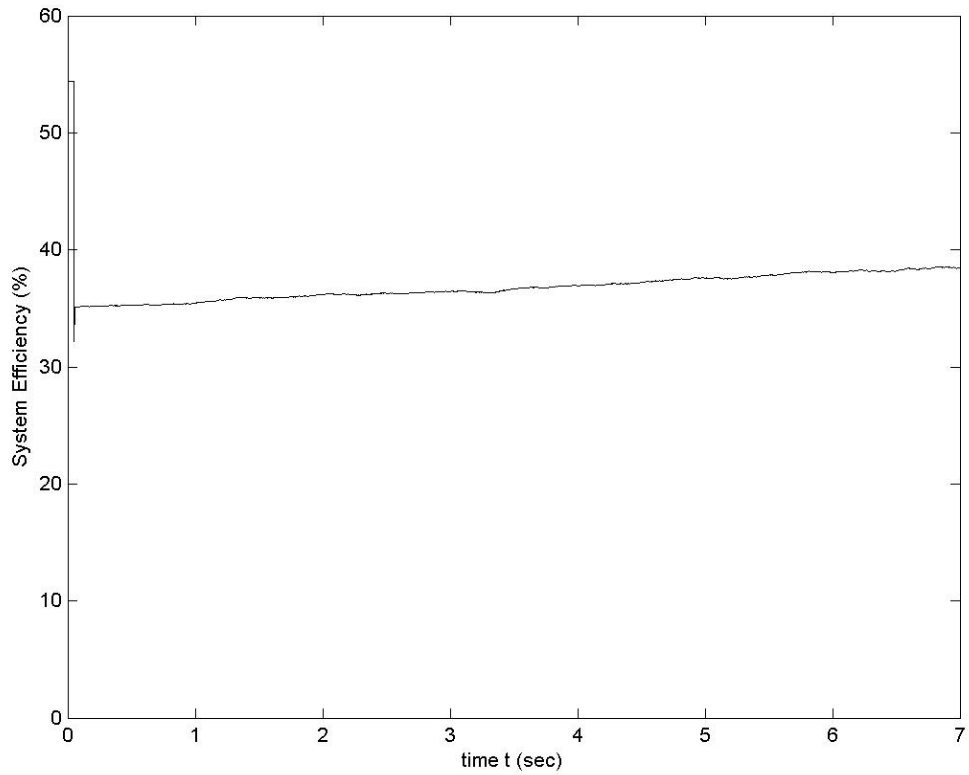


Figure 26: Simulated Pressure Drop Across Hydraulic Motor versus Time

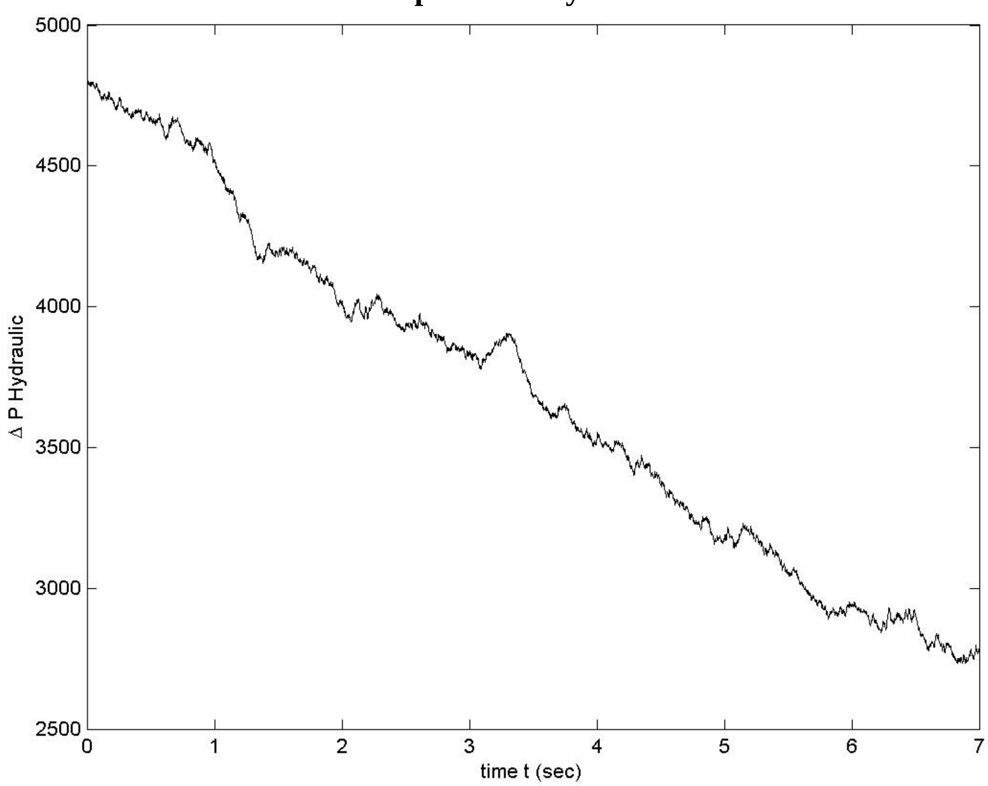


Figure 27: Estimated Hydraulic Motor Efficiency versus Displacement

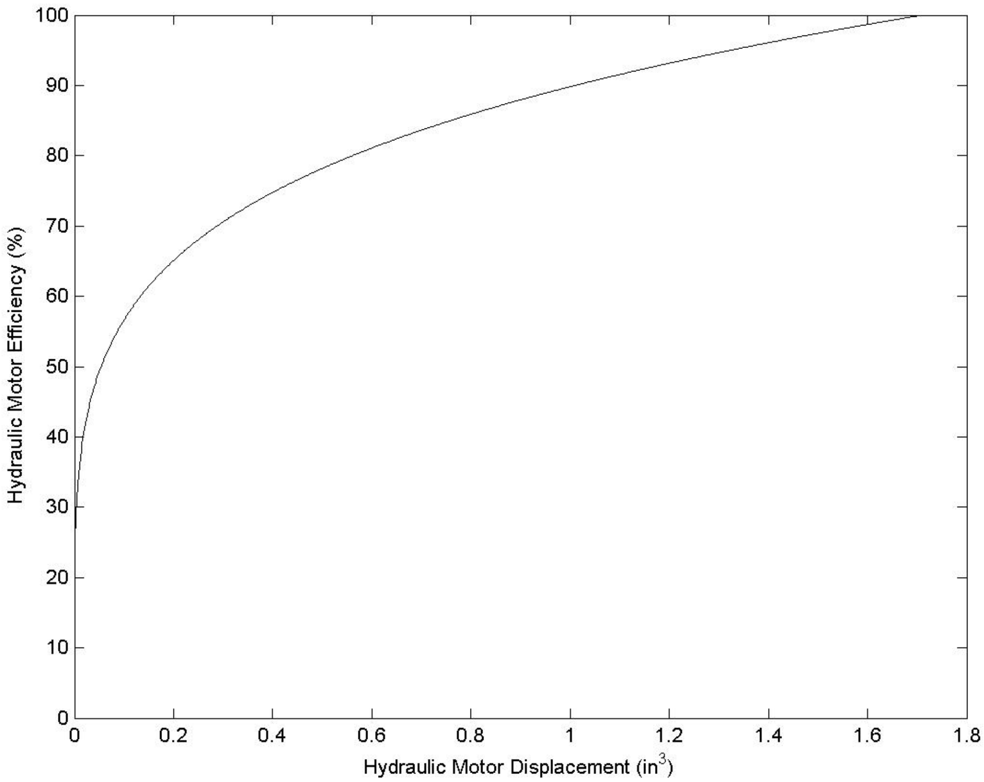


Figure 28: Hydraulic Motor Efficiency versus Time

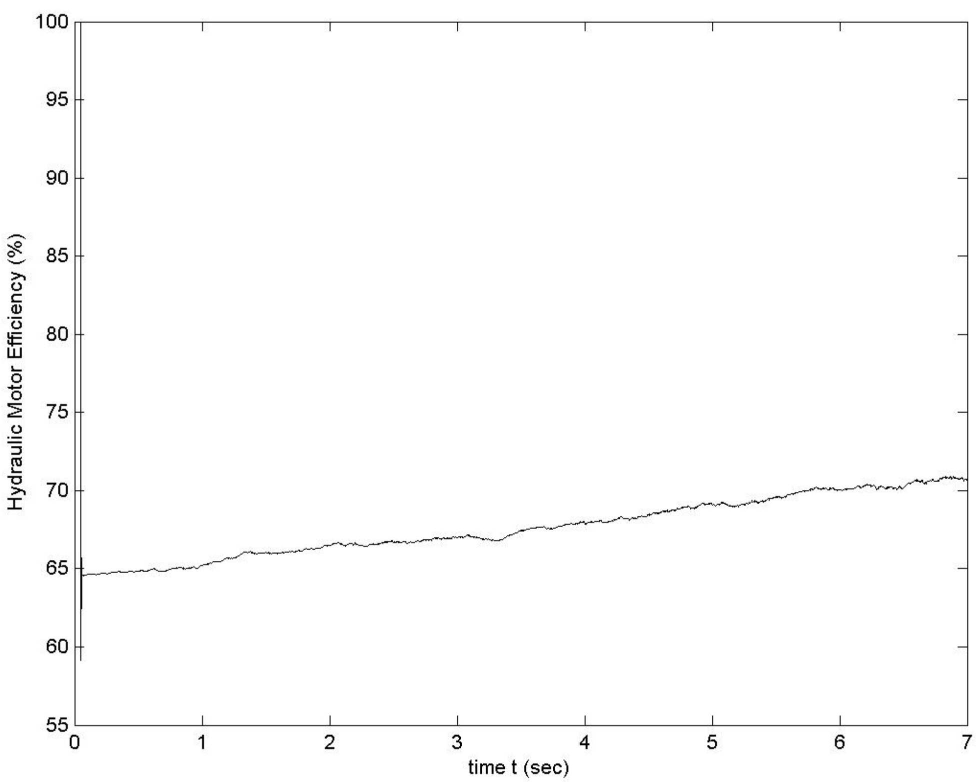


Figure 29: Supercharger RPM versus Time

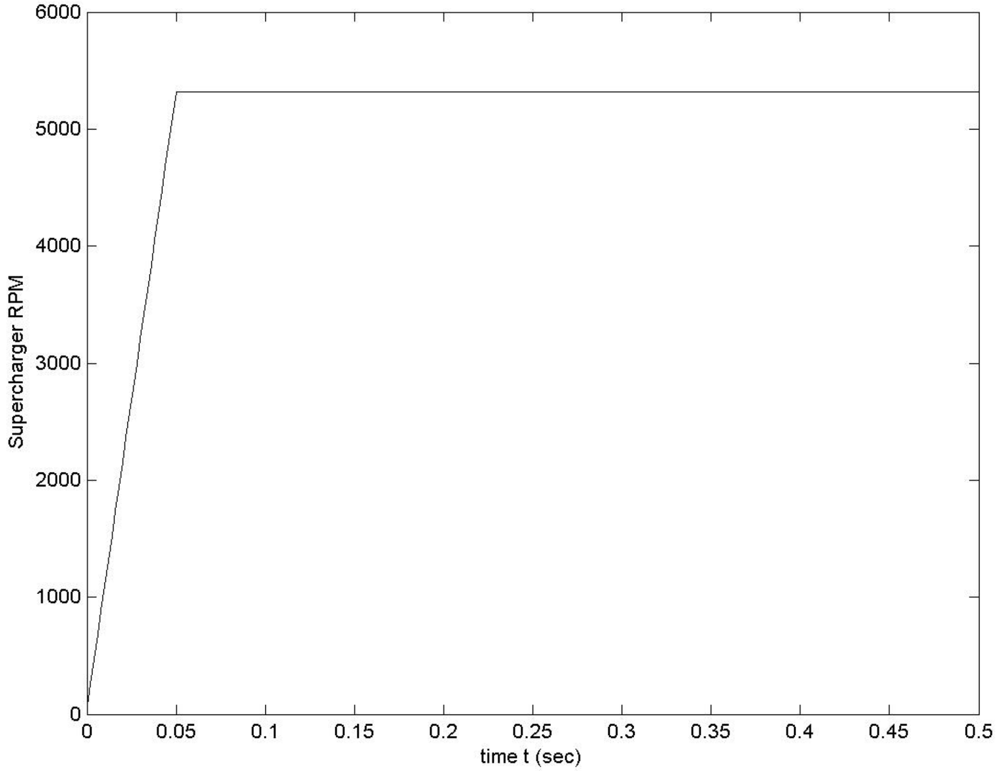


Figure 30: Hydraulic Motor Displacement versus Time