ABSTRACT
Under the hood valve systems for heavy diesel applications often rely on electrical or hydraulic systems to actuate. Imagine if you could use an existing media on-board that was intrinsically safe and environmentally friendly. A poppet style pneumatic valve powered by a voice coil actuator would yield a highly accurate, robust and efficient option to traditional actuation systems.

INTRODUCTION
A traditional valve system consists of direct electrical and electro-hydraulic actuators. Direct electrically driven valves require electric motors to create a needed force. This force is created by the direct conversion of electric current into torque. These motors often have a service rating for temperatures up to 200°F (93.3°C). The low temperature rating is the limiting factor for most motors because of their permanent magnets and electrical solder joints. Magnets and high temperature solder exist but are not classically used due to cost constraints. These limitations cause difficulty in under-hood applications where the typical ambient temperature exceeds 300°F and per unit cost is a high visibility consideration.

Electro-hydraulic valves commonly utilize engine lube oil coupled with a solenoid operator to create a required force. This style of valve is much more durable in applications exceeding 300°F (149°C). This is due to the ability to insulate the electrical components from the heat source. Some of the limiting factors for a classical electro-hydraulic valve are flammable working media and a relatively short working stroke. Solenoid force decreases exponentially with increasing stroke. If longer stroke is required a solenoid must drive a secondary valve to operate the primary system.

Currently many pressure control valves exist on the market today. One type is a spool type valve. This type uses a tight fitting spool in a precision honed bore. Spool valves are extremely sensitive to contamination in the working media. They also work poorly with low viscosity fluids such as air, due to inherent gap between the components.

Poppet style pressure control valves have many benefits when compared to spool types. Poppet style valves have very low contamination susceptibility, low leakage and the ability to withstand high pressure operation. A typical poppet style version utilizes two solenoid operated valves. This is because the design allows for one valve to reduce free stream pressure at a certain point and the other valve to relief pressure from that point. This is undesirable because of the increased complexity of the system.

APPLICATION AND REQUIREMENTS:
The mission of the design team was to develop a pressure control valve for a pneumatic application. The poppet style valve was chosen due the qualitative aspects. As for operating the valve, a voice coil operator was chosen for its high force, high accuracy and low hysteresis.

The pressure control valve will be used primarily in heavy diesel truck applications in which air is the working media. The end users of the valve specify the following requirements for heavy diesel truck applications. The valve must endure temperature extremes form -40°F to +450°F (-40°C to 232°C) and supply a pressure range of 10psi to 90psi (69kPa to 620kPa) with a response time of 50ms. The poppet style valve coupled with a voice coil operator will accomplish these requirements.

BASIC CONCEPT AND OPERATION:
A Poppet style valve is used in industry for pneumatic applications due to their low contamination susceptibility and low degree of internal leakage. However most pressure control poppet valves on the market utilize two separate actuators, effectively two
Valves are used to control pressure at one point. Our design is an attempt to use only one actuator to control two poppets using a creative means of force balancing. The actuator of choice is a voice coil which will give proportionality and zero magnetic hysteresis. Before designing the voice coil, the valve section must be designed to determine the proper force needed. Figure 1 illustrates the valve design and the common parts.

Figure 1. Valve cross section

When charging, the relief poppet is pushed by the voice coil (not shown in Fig. 1), which comes in contact with the relief seat. The relief and reducing poppet can be viewed as one mass during the charging of the valve. The force from the voice coil now acts on the reducing poppet. The reducing poppet lifts off the reducing seat allowing airflow into the working port. The design will use a spring in both the valve and voice coil sections so the unit will have a preferred orientation which is normally closed in case of system failure.

When discharging, the force on the relief poppet is reduced by decreasing the current to the voice coil. This causes a force imbalance between the relief seat and the reducing seat. The pressure in the working port acts on all surfaces between the relief and reducing poppets. Spring #1 and the pressure in the common port cause the airflow area to reduce, allowing pressure to increase or decrease accordingly. The relief seat remains in contact with the relief poppet not allowing premature venting to occur. After the reducing poppet contacts the reducing seat, the relief poppet lifts off the relief seat causing the valve to vent. The venting is controlled by the force the voice coil exerts on the relief poppet. The desired pressure is achieved by controlling the current to the voice coil actuator.

Figure 2. Note the slight difference between the charging and discharging curves. The expected hysteresis in this design is inherent to the springs chosen. The largest effect on this band is seen in the preload chosen for spring #1.

**ORIFICE SIZING FOR RESPONSE**

The first thing to consider is sizing the valve for the required response time. From the requirements listed above we can determine the flow rate to charge the given volume in the allotted time. This is achieved by the definition of flow rate as some volume per unit time.

\[ Q = \frac{\text{V}}{t} \]  \hspace{1cm} (1)

We also know that flow rate can be defined by the velocity of a fluid flowing through a specific cross sectional area.

\[ Q = V \times A \] \hspace{1cm} (2)

The orifice size is relevant to the flow area when the valve is open for charging. In order to find the required orifice size, the area from Eq. (2) is required. In order to calculate the area, velocity and flow rate need to be determined. The following equations are defined and simplified for use. Equation (3) represents the Ideal Gas Law and Eq. (4) is the geometric definition of a circle. Equation (5) represents Bernoulli’s equation.

\[ \rho = \frac{P}{RT} \] \hspace{1cm} (3)
The following assumptions are made to simplify Bernoulli’s equation:
- Charging and Discharging require the same orifice size.
- Steady Compressible frictionless flow.
- Neglect Heat Transfer.
- Air behaves like an ideal gas.

Using Bernoulli’s equation and simplifying Eq. (5) based on our assumptions.

\[ \frac{2}{1} \frac{dV}{dt} - \frac{V^2}{2} = 0 \]  
(6)

Substituting Eq. (3) into Eq. (6)

\[ \frac{2}{1} \frac{RT}{P} dP - \frac{V^2}{2} = 0 \]  
(7)

Rearranging Eq. (7)

\[ V_1^2 = 2RT \frac{1}{P} dP \]  
(8)

Then integrating Eq. (8) we get.

\[ V_1^2 = 2RT \ln \frac{P_2}{P_1} \]  
(9)

To find \( V \) we can model a static volume with a charged pipe flowing into it. See Fig. 3. The pressures in Fig. 3 were selected to represent the case where the velocity will be the slowest at the extreme of the valves operating range.

**SEAT DIAMETER SELECTION**

The main driving factors concerning seat diameter are the selection of stock diaphragms, charging pressure of the valve and the desired output force of voice coil. Since the valve will be charging to 90psi (621kPa) and the desired output of the voice coil should be less than 10lbs (44.5N), these values translate to a diameter of less than 0.35in (0.89 cm) or area of less than 0.1 in\(^2\) (0.65 cm\(^2\)). Based on this, the only diaphragm in the DIA-COM catalog that meets our requirements is FC-38-12.
Specifications:
- Cylinder Diameter: 0.380 in (0.97 cm).
- Piston Diameter: 0.250 in (0.64 cm).
- Pressure Area: 0.07 in² (0.45 cm²).
- Material: Nitrile Rubber

Since this part of the valve needs to be sized so that it is pressure balanced, we need to have equal pressure areas on both sides. Therefore, we need to size the seat diameter such that it matches the area of the diaphragm. Using equation 4.

\[
.07\text{in}^2 = \frac{\pi * d_s^2}{4}
\]

\[
d_s = \sqrt{\frac{4 * .07\text{in}^2}{\pi}}
\]

\[
d_s = .2985\text{in} \quad (0.758 \text{cm}) \quad \text{Solution 5}
\]

This also gives us the maximum pressure force that will act on the face of the poppet.

\[
F = PA
\]

\[
F = 100\text{ psi} * .07\text{in}^2
\]

\[
F = 7\text{ lbs} \quad (31.1\text{ N}) \quad \text{Solution 6}
\]

This is the primary force the voice coil must overcome.

**POPPET CRITICAL FEATURE SIZING**

We need to make certain when the poppet is fully opened we have an appropriate annular flow area to ensure the valves response time. We know the required flow area from above is \(A=.02139\text{in}^2\) (0.14 cm²). We also need to guarantee the poppet shaft wall thickness is reasonably thick, for machine-ability not structural.

From Fig. 4 we see that \(D_c\) and \(D_t\) are already known, so we only need to explore \(D_i\). To find \(D_i\) we need to determine the annular area, or area between two concentric circles.

\[
A = \frac{\pi D^2}{4} - \frac{\pi D_i^2}{4} = \frac{\pi (D_s^2 - D_i^2)}{4}
\]

Rearranging Eq.1.12 and solving for \(D_i\).

\[
D_i = \sqrt{D_s^2 - \frac{4A}{\pi}}
\]

Solving for \(D_i\).

\[
D_i = .249\text{in} \quad (0.63 \text{cm}) \quad \text{Solution 7}
\]

The values calculated express the extreme where lowest flow is allowable. Tolerance will be added so that these dimensions become a high or low limit.

**POPPET STROKE DETERMINATION**

The required stroke is determined by the interface between the poppet and the seat.

The valve stroke must allow a minimum curtain area for the valve to meet flow requirements. The curtain area is defined as the area between the poppet and the seat while the poppet is in the flow field. To achieve full flow, the curtain area must meet or exceed the seat inlet area. To determine the curtain area we will approximate its area as the perpendicular distance from the poppet surface to the seat multiplied by the circumference of the seat.
\[ A = \pi D^2 b \]  
(12)

Rearranging and solving for \( b \).

\[ b = \frac{A}{\pi D} \]

\[ b = 0.0228 \text{ in} \ (0.058 \text{ cm}) \quad \text{Solution 8} \]

Based on the trigonometric definition of a triangle we can determine the required stroke.

\[ \cos\left(\frac{\alpha}{2}\right) = \frac{b}{\Delta x} \]  
(13)

Rearranging and solving for \( \Delta x \).

\[ \Delta x = \frac{b}{\cos\left(\frac{\alpha}{2}\right)} \]

\[ \Delta x = 0.0322 \text{ in} \ (0.082 \text{ cm}) \quad \text{Solution 9} \]

The voice coil will move in two directions for charging and discharging therefore the total stroke for the voice coil is \( 2\Delta x \).

**VOICE COIL.**

The basic principle shown in Fig. 9 states when current flows in a conducting wire, which is in a magnetic field, a linear force results.

\[ F_B = qv \times B \]  
(14)

This is the primary theory used to design a voice coil. Figure 6 describes the theory in Cartesian coordinates. A voice coil is the application of this theory in radial coordinates. It can be equated to a wound coil with magnetic poles through the center. This would also result in a linear force. The direction of movement is dependant on the direction of current flow.

We define \( F_B \) as the magnetic force, \( q \) is the charge on the particle, \( v \) is the velocity vector of the charged particle, and \( B \) is the magnetic field vector or flux density. When the angle between the velocity vector and the magnetic field vector is not equal to zero the magnetic force will act in a direction perpendicular to the plane formed by \( v \) and \( B \).

\[ F_B = \mathbf{I} \times \mathbf{B} \]  
(15)

We define \( I \) as the charge moving through a conducting wire, \( L \) as a vector in the direction of the current and has a magnitude equal to the wire length. The cross product of \( L \) and \( B \) results in their magnitudes being multiplied. This is due to the vectors being perpendicular.

In our case with a voice coil, the current is carried through a closed loop in a uniform magnetic field. Therefore we can express Eq. (15) in a new form, which takes the vector sum of the length elements \( ds \) over the entire loop.

\[ F_B = I \left( \int ds \right) \times B \]  
(16)

The end result yields an equation that is easily used to determine the force a voice coil will produce. This equation can be simplified further. This can be done by assuming the magnetic flux is linear in the radial direction from north to south across the mean diameter of the wound coil radial and the B-H curve of a given magnet becomes linear as shown in Fig. 10. (Where \( B \) is the flux density and \( H \) is the coercive force.)

![B-H curve](image)

Figure 6. Basic theory of current carrying wire in a magnetic field.

Figure 7. B-H Curve
These assumptions will result in the integration in Eq. (16) simply becoming the length of the current carrying wire. We also need to take into account that this system is not an ideal case. There are losses that need to be taken into account. Losses in the system are due to the linear assumption of the B-H curve, variations in the static magnetic field, flux leakage, changing resistance due to heating and changing inductance. The result is a simplified linear equation with a correction factor $K$.

$$F_B = KBLI$$

(17)

To determine the correction factor several assumptions must be made. First, no flux leakage occurs through the entire system, which is for an ideal case. Second, the length of the magnet to be much greater than the length of the gap between the magnets surface to the mean diameter of the coil. Finally, the field strength of the magnet is equal to the field strength of the gap. The correction factor was based on empirical data from testing of a known voice coil. Also, the correction factor for this coil will be different than our coil. Therefore it is assumed that we can derive an additional correction factor from the existing coil to correlate to ours.

Testing consisted of applying current to the coil and measuring the force output using a load cell. Table 1 shows the empirical data of the test sample from our sponsor.

<table>
<thead>
<tr>
<th>Force [lbs]</th>
<th>Force [N]</th>
<th>I [Amps]</th>
</tr>
</thead>
<tbody>
<tr>
<td>0</td>
<td>0</td>
<td>0</td>
</tr>
<tr>
<td>0.285</td>
<td>1.27</td>
<td>0.153</td>
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<td>0.45</td>
<td>2.00</td>
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<td>0.57</td>
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<tr>
<td>0.734</td>
<td>3.26</td>
<td>0.31</td>
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<tr>
<td>1.004</td>
<td>4.47</td>
<td>0.409</td>
</tr>
<tr>
<td>1.236</td>
<td>5.50</td>
<td>0.5</td>
</tr>
</tbody>
</table>

Table 1. Force versus current.

Using techniques developed by the design team with aid of the sponsor, a correction factor was determined. The value and method of this calculation is unique to this coil.

**SIZING AND COMPONENTS**

The two major components in the design of a voice coil are the magnets and gage of wire. These are chosen by our package limitations and the flux density of the magnets. The wire size selected was #24 AWG copper wires. The magnets selected are SmCo (Samarium Cobalt) with strength of N28. The sizing of the coil is done by first determining the overall diameter of the package and then finding the corresponding magnets to fit. After determining the magnet size we can optimize the coil by varying the wire gauge and length to achieve the required number of turns and resistance of the coil. The wire diameter affects the resistance as well as the overall diameter, wire length, number of turns and turns per layer. The force required for the coil was determined from the valve section to be 7.5lbs (33.4 N), which does not include forces from the valve spring. Total force for the voice coil to overcome is 9.5lbs (42.3 N). Optimization was achieved by using Ohms law [Eq. (18)] and the required amp turns determined from Eq. (17). Amp turns are the product of number of turns of wire in the coil (total length of wire) and the current applied to the coil. Table 2 shows the results of the optimization.

$$\text{Ohms Law: } V = IR$$

(18)

<table>
<thead>
<tr>
<th>Coil Sizing</th>
<th></th>
<th></th>
<th></th>
<th></th>
<th></th>
<th></th>
<th></th>
<th></th>
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<tbody>
<tr>
<td>Wire Gauge</td>
<td>24 AWG</td>
<td>Wire Diameter</td>
<td>0.0201 in</td>
<td>0.051 cm</td>
<td></td>
<td></td>
<td></td>
<td></td>
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<tr>
<td>Resistance</td>
<td>25.67 Ohms</td>
<td>Length of Wire</td>
<td>2824 in</td>
<td>7173 cm</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Total Resistance</td>
<td>6.04 Ohms</td>
<td>Layer OD</td>
<td>1.59 in</td>
<td>4.03 cm</td>
<td></td>
<td></td>
<td></td>
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</tr>
<tr>
<td>Amp Turns/Layer</td>
<td>41</td>
<td>Turns/Layer</td>
<td>17</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

Table 2. Coil sizing chart.

The result of optimization and Eq. (17) we can plot force versus current shown in Fig. 11.
MATERIAL SELECTION

Materials were selected for their strength, corrosion resistance, ease of machining and thermal properties.

Aluminum was selected due to its low density, therefore being light weight. 6061-T8 was selected for high resistance to corrosion and cracking, joining characteristics and ability to accept applied coatings (i.e. anodize). 6061-T8’s yield strength is greater than most of the steels selected because the aluminum is a major structural member of the valve. The cartridge body and all non functional parts were made from aluminum.

Stainless steel was selected due to extremely high corrosion resistance. AISI type 303 stainless steel was chosen for machineability, minimal magnetic properties and availability. The 303 Stainless Steel parts provide structural members for the assembly and undergo no loading other than thermal loading. Any non-magnetic stainless steel could be selected for these components. The outer housing for the voice coil was made from AISI 303 Stainless Steel.

420F and 440C stainless steels possess excellent corrosion resistance, machineability and ability to be hardened with heat treatment. The poppets are made of 440C and seats of 420F. They need to resist corrosion while being able to have different hardness’s for sealing when seated. Generally, we would heat treat the poppets to a hardness (Rockwell C scale) 2 to 3 points harder than the seats. This will cause a slight deformation of the seat as the poppet contacts it. This is favorable due to the elimination of any leak paths from the materials permanent deformation onto each other.

PRELIMINARY TEST RESULTS

All testing was done at G.W. Lisk Co., Inc. per the test plan TPM2-3021. The valve was tested using a solenoid operator. At the time of this test, the voice coil operator was not complete. Testing was done using the following components.

- Sensotec Pressure Transducer.  
  Model # THE/708-12
- GW INSTEK DC Power Supply.  
  Model # SPS-3610
- DAYTRONIC Strain Gage Conditioner.  
  Model # 3270
- YOKOGAWA Digital Oscilloscope.  
  Model # DL1540
- G.W. Lisk Labview equipped test stand.

Response time testing was done at 15 PSIG inlet and 50 PSIG inlet. The solenoid actuator had a response time of approximately 70ms for the coil inductance. The system response time for both 15 and 50 PSIG were both approximately 235ms to achieve full charged pressure. The response time of the solenoid alone is greater than the requirements per our sponsor.

Figure 11. 15PSIG Response Time
Pressure versus current testing was also completed. The valve was actuated using a solenoid operator. Performance curves are shown below.

The performance curves prove that the restoring force of the spring in the valve is the primary driving force in the hysteresis loop. Higher pressure effectively allow us to neglect the force of the spring acting on the reducing poppet.

ACKNOWLEDGMENTS
We would like to thank everyone who made this project possible.

G.W. LISK VALVE ENGINEERING AND PROTOTYPE DEPARTMENTS.
Gary Garcia
Mark Sahler
Scott Twitchel
Greg Bree
And the members of the Prototype Machining Department.

SUPPLIERS
TRIDUS Magnetics
F&S Distributors Inc.
Lee Spring Co.
Dia-Com

LIST OF REFERENCES.


