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SUMMARY

This report details the design of a buoyancy engine for use in a shallow water underwater glider AUV. There is a brief introduction to underwater gliders and buoyancy engines then five possible designs that are candidates for a suitable buoyancy engine are presented. These five candidates are then evaluated using a mathematical model to examine their system dynamics. A cost and energy analysis is also performed to help in evaluating which of the five candidates was most suitable. After the evaluation the structural layout for the best candidate will be detailed.

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Summary

This report details the design of a buoyancy engine for use in an underwater glider. A brief introduction to underwater gliders and buoyancy engines is presented first, followed by several opposing design constraints. Then five different design candidates are presented, which are evaluated using a mathematical model of the buoyancy engine in order to examine the system dynamics. Cost and energy analyses are also conducted for the candidates. Upon evaluating the five candidates the one that is most suitable for our purposes will be designed. The design section of the candidate chosen will detail its structural layout. Conclusions and future work follow, which summarize some important things covered in this report as well as lay a path for the further development of the buoyancy engine.

Table Of Contents **Page**

List of Figures **Page**

List of Tables **Page**

Nomenclature

Symbol	Description
$F_{buoyant}$	Buoyant force acting on cylinder. [N]
F_{drag}	Hydrodynamic drag force on cylindrical body. [N]
W	Weight of cylinder in air. [N]
M	Apparent mass of the cylinder. [kg]
$\ddot{z}$	Acceleration of the cylinder in the z direction. [m/s^2]
$\dot{z}$	Velocity of the cylinder in the z direction. [m/s]
z	Position of the cylinder in the z direction. [m]
ρ_w	Density of water. [kg/m^3]
g	Acceleration due to gravity. [m/s^2]
V	Volume displaced by the cylinder. [m^3]
A_o	Cross sectional area of the cylinder. [m^2]
c_d	Drag coefficient for cylinder in axial flow.
d	Diameter of the cylinder. [m]
m	Mass of the cylinder. [kg]
m'	Added mass due to cylinder acceleration. [kg]
$V_{cylinder}$	Volume of the cylinder at neutral buoyancy. [m^3]
ΔV	Change in cylinder volume. [m^3]
$\dot{z}_{Term}$	Cylinder terminal velocity in the z direction. [m/s]
p	Percentage of the cylinder's terminal velocity.
P_{depth}	Pressure at a given depth. [Pa]
h	Measure of depth. [m]
F_{Total}	Total force acting on actuator. [N]
ΔP	Difference between internal and external pressure. [Pa]
A_{diap}	Effective pressure area of diaphragm. [m^2]
P_{atm}	Atmospheric pressure. [Pa]
P_{vacuum}	Pressure inside cylinder due to vacuum. [Pa]
P_{diff}	Differential pressure acting on the piston. [Pa]
A_{Piston}	Effective pressure area of piston. [m^2]
$d_{Cylinder}$	Diameter of internal hydraulic cylinder. [m]
L_{Stroke}	Linear actuator stroke. [m]
$T_{\Delta V}$	Time taken to displace ΔV. [s]
C	Volumetric flow rate. [m^3/s]

Table of Constants

Symbol	Description	Value
g	Acceleration due to gravity.	9.81 [m/s^2]
π	pi.	3.14
c_d	Drag coefficient for blunt end cylinder in axial flow with 10:1 length to diameter ratio.	0.82
P_{atm}	Atmospheric pressure.	101325 [Pa]
ρ_w	Density of water.	1000 [kg/m^3]
ρ_{pvc}	Density of PVC.	1013 [kg/m^3]
ρ_{al}	Density of aluminum.	2700 [kg/m^3]
ρ_{st}	Density of steel.	7850 [kg/m^3]

1.0 Underwater Gliders

Underwater gliders are relatively new experimental autonomous underwater vehicles (AUVs) that rely on changes in buoyancy to operate. They are identified by their buoyancy engines, fixed wings, streamlined bodies, internal mass repositioning, and lack of external mechanical parts. Many underwater gliders contain computerized control systems, sensory equipment, satellite communications and GPS receivers. They can be programmed to dive to certain depths and follow waypoints while collecting data, then glide back to the water surface, get their bearings from GPS and relay information gathered via radio or satellite communication links.

Buoyancy engines in underwater gliders change the mass or the volume of the vehicle and therefore control the net buoyant force acting on the vehicle. By making the underwater glider negatively buoyant the vehicle descends, water flowing over the wings creates life and drag which push the glider forward during its downward motion. Some gliders move internal masses inside their hulls (such as batteries) with an electrical motor to adjust the pitch angle of the vehicle in order to achieve desired glide angles. Upon reaching a target depth the buoyancy engine is used to make the underwater glider positively buoyant, while the vehicle ascends the water flow over the wings give the glider forward momentum. This “sawtooth” movement in the vertical plane is the general motion trajectory followed by underwater gliders, which is characterized by the gliding dive (negative buoyancy, pitched downwards) followed by the upward glide (positive buoyancy, pitched upwards). To move from side to side, underwater gliders either make use of a rudder to adjust heading, or shift some internal mass to bank similar to an airplane.

Underwater gliders generally have long range and high endurance. Energy consumption rates are designed to be as low as possible to ensure maximum operational time. Since higher speeds create more hydrodynamic drag, gliders tend to move at lower speeds, which grants them higher endurance. Typical gliders run at half a knot on half a Watt.

Several underwater gliders exist today, some of which can be purchased. The Slocum Glider is an example of one such glider (Figure 1). It was developed at the Webb Research Corporation in East Falmouth, Mass. There are presently two versions of the

Slocum glider, “Coastal” and “Deep Ocean”. The Coastal Slocum glider relies on a conventional hydraulic pump engine, driven by an electrical motor and batteries as its buoyancy engine. The hydraulic pump transfers oil between an internal and external reservoir that leads to change in buoyancy. For navigation it uses GPS and dead reckoning. This version of Slocum glider has an endurance time of 15 to 30 days, can dive as deep as 200 metres, and sells on the market for $100,000.

The Deep Ocean Slocum glider relies on a “Thermal Engine” to control buoyancy. It has a typical endurance time of 5 years, and is capable of depths up to 1200 metres. Again, for navigation the Deep Ocean Slocum glider relies on GPS and dead reckoning.

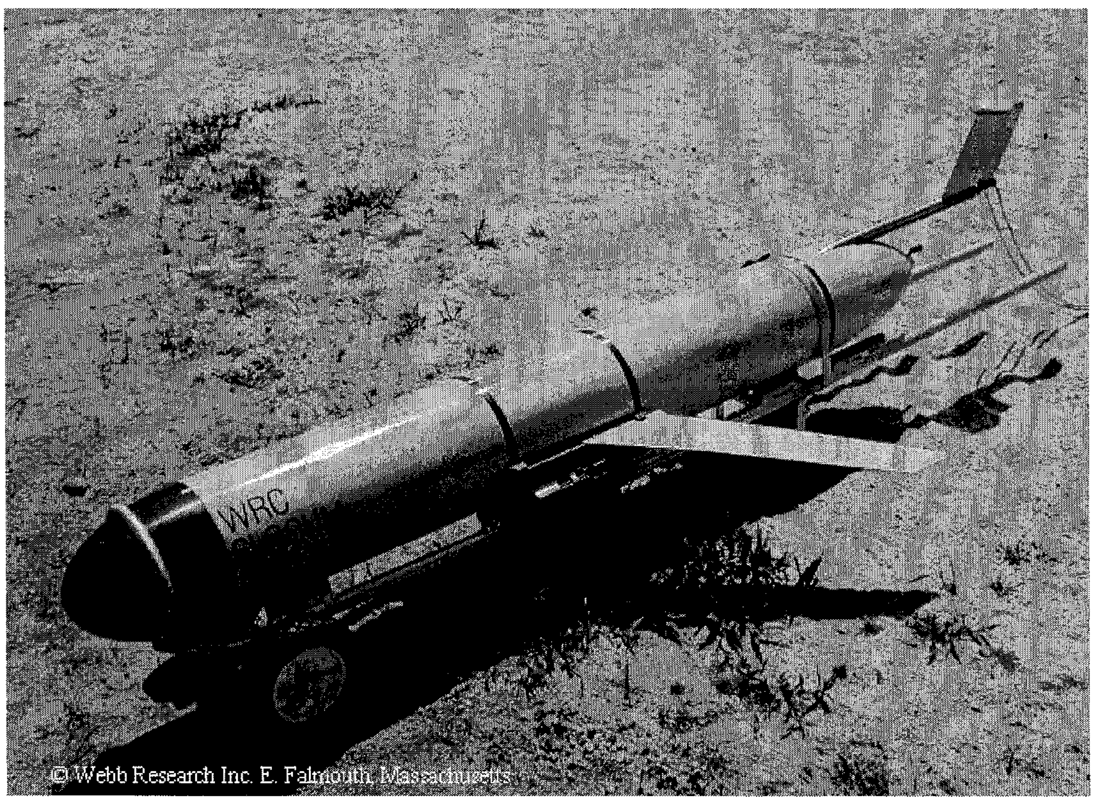


Figure 1 *– Slocum Glider, Webb Research Corporation.*

Seaglider, developed at the University of Washington, is another example of an existing underwater glider. It has a stable, low-drag, hydrodynamic shape (Figure 2). Seaglider has an endurance time of 6 months and is capable of depths up to 1000 metres, the hull compresses as it sinks, matching the compressibility of seawater. It can travel on slopes as gentle as 1:5 or as steep as 1:3. Seaglider relies on GPS and dead reckoning for navigation and satellite for communications.

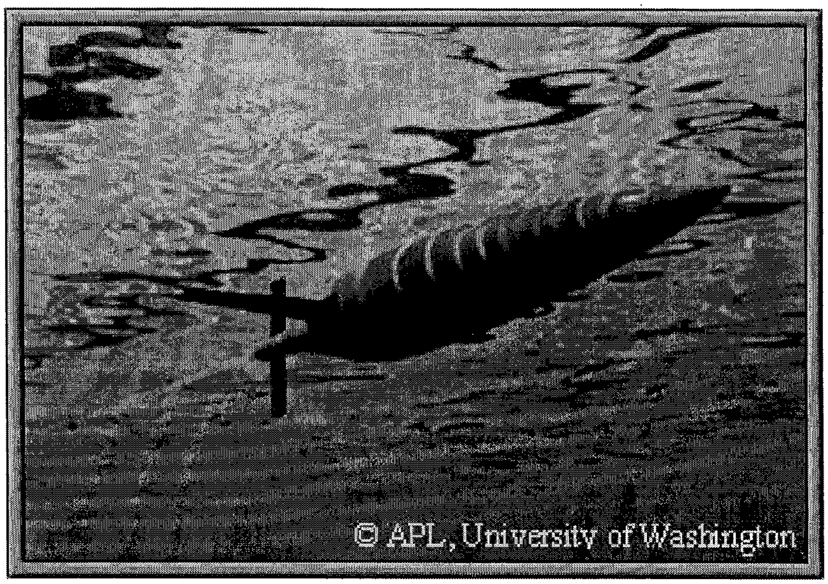


Figure 2 *– Seaglider, University of Washington.*

There are hundreds of applications for underwater gliders. Some examples include collecting temperature and salinity measurements, measuring currents and eddies, profiling the ocean floor, counting microscopic plants and even recording biological sounds (whale songs). The US Navy intends to utilize underwater gliders to check for mines and other such countermeasures in foreign waters.

2.0 Buoyancy Engines

A buoyancy engine controls the magnitude and direction of the buoyant force acting on a submersed body by changing the volume or weight of the body. There are several methods used presently to change the buoyant force on a body. In one variation a pump is used to transfer oil between an internal and external reservoir, therefore varying the volume of the submersed body, which leads to a change in buoyancy. The same effect can be achieved using a hydraulic or electromechanical linear actuator and piston assembly to transfer oil between an internal cylinder and external reservoir. Another variation involves using a hydraulic or electromechanical linear actuator to expand and retract a rolling diaphragm attached to the body; this changes the volume of the body that then leads to buoyancy change. An external pump transferring surrounding fluid between the surroundings and a reservoir inside the body is also a functional buoyancy engine. In designing the buoyancy engine for use in an underwater glider we have chosen to focus only on the vertical motion the buoyancy engine is required to produce. The buoyancy engine will be installed in a 10 cm diameter PVC pipe of 1 m length, which is about two-thirds the size of the Slocum Glider. The pipe will travel up and down while oriented with its longitudinal axis vertical in the water. The testing tank at IOT is only 7 metres in depth but the buoyancy engine will be designed to travel as deep as 20 metres, which is a 2.56 safety factor.

3.0 Conceptual Thinking

3.1 Design Constraints

Before introducing the conceptual designs of several buoyancy engines, a discussion on constraints is necessary. There are several design constraints described in this section. Each constraint applies limitations on the designs to be proposed. The constraints are:

Size

The buoyancy engine will be used in a standard 10 cm diameter 1 m long piece of PVC pipe. This has been chosen as a scaled down version of the Slocum Glider AUV, which also has a 10:1 length to diameter ratio. This size constraint limits the size of parts that the buoyancy engine uses to operate. Components such as pumps, linear actuators, and electronics must all be able to fit inside the pipe.

20 Metre Depth

The buoyancy engine should be designed to operate to a depth of 20 metres. This will limit all designs due to the pressure imposed on the buoyancy engine at the maximum depth. Pumps must have suitable differential pressure rating and linear actuators must supply enough force when subjected to the pressure experienced at a depth of 20 metres.

3.2 Design Candidates

Different buoyancy engines will be introduced briefly in this section. The buoyancy engines will later be evaluated to determine which of those presented is the best candidate to be designed.

3.2.1 Candidate #1

This system involves using a bi-directional pump to transfer a fluid such as air or oil between an internal and external bladder (Figure 3). When fluid is transferred from the internal bladder to the external bladder, the submersed body occupies more volume; therefore the buoyant force acting on the body increases. When the fluid is transferred from the external bladder to the internal bladder, the submersed volume decreases causing a decrease in the buoyant force.

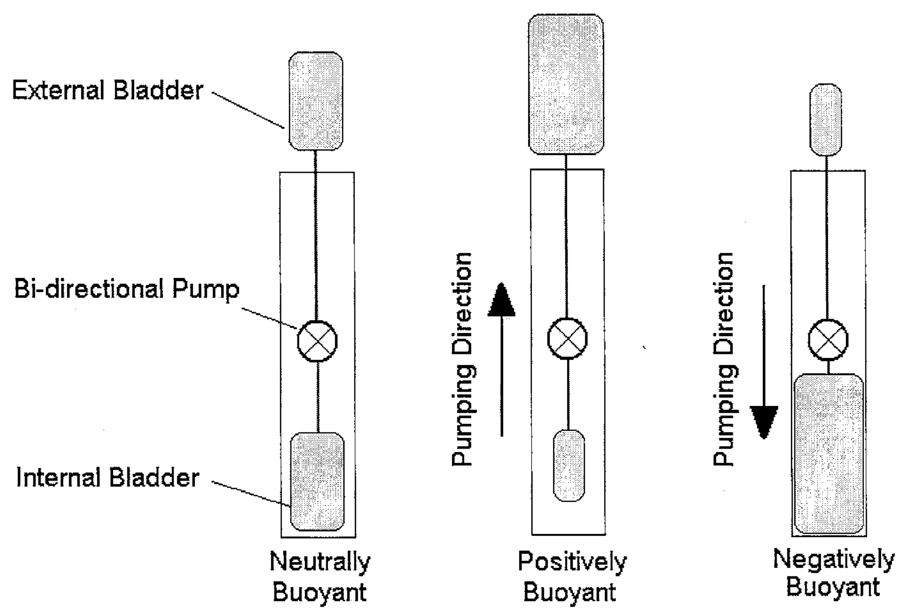


Figure 3 *– Conceptual Drawing of Candidate #1.*

3.2.2 Candidate #2

A uni-directional pump may also be used when combined with the proper flow direction switching valves. The valves would typically be 12 to 24 volt solenoid activated, and would simply switch the flow direction (Figure 4). This system works identical to the bi-directional pumping setup, with the exception that switching the flow direction requires energizing or de-energizing the solenoid valve. Transferring oil between external and internal bladder causes volumetric change, therefore the buoyant force changes.

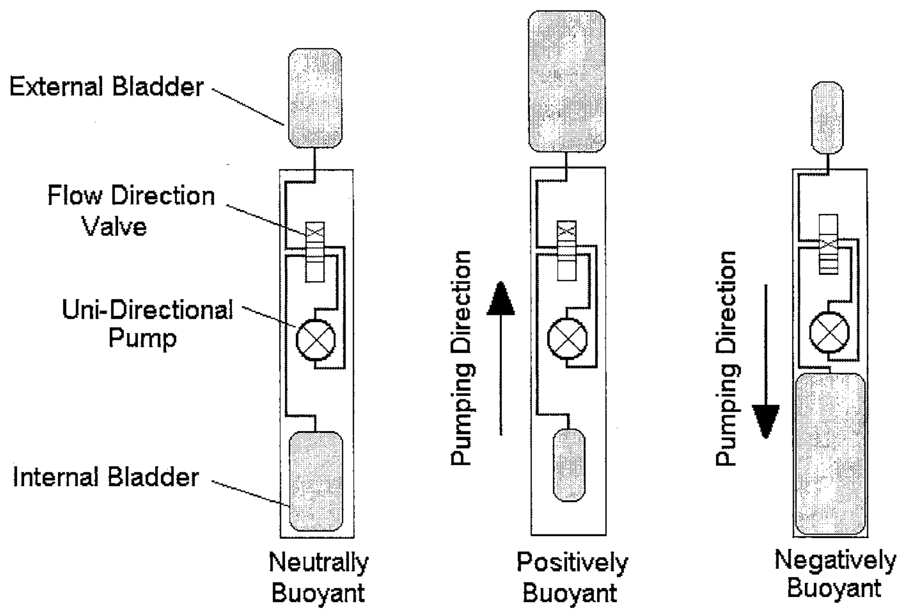


Figure 4 – *Conceptual Drawing of Candidate #2.*

3.2.3 Candidate #3

This method involves the use of a linear actuator and a rolling diaphragm. When the linear actuator extends or contracts the rolling diaphragm deforms and the volume of the body increases or decreases causing change in buoyancy (Figure 5).

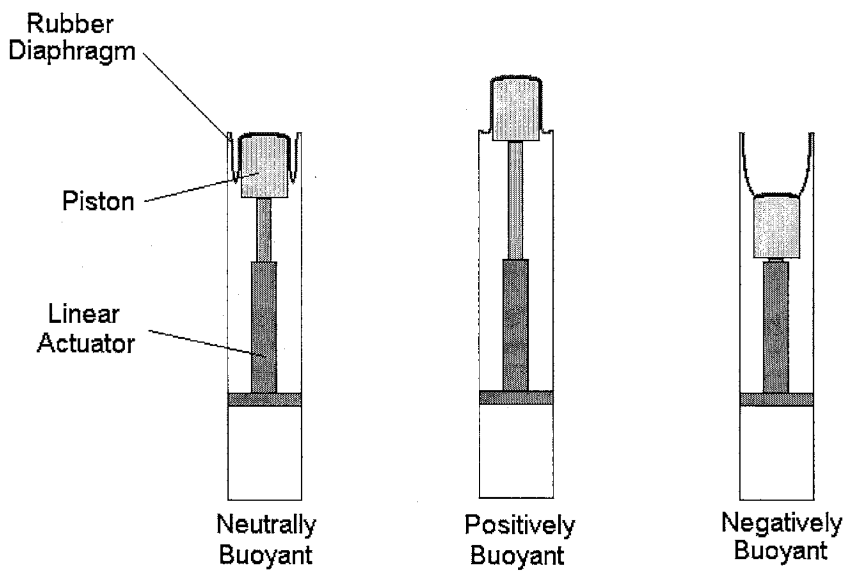


Figure 5 *– Conceptual Drawing of Candidate #3.*

3.2.4 Candidate #4

This buoyancy engine design combines a linear actuator with an external bladder. The linear actuator moves a piston inside an internal cylinder, which in turn transfers working fluid to and from the external bladder (Figure 6). The fluid entering or leaving the external bladder changes the volume and therefore changes the buoyant force.

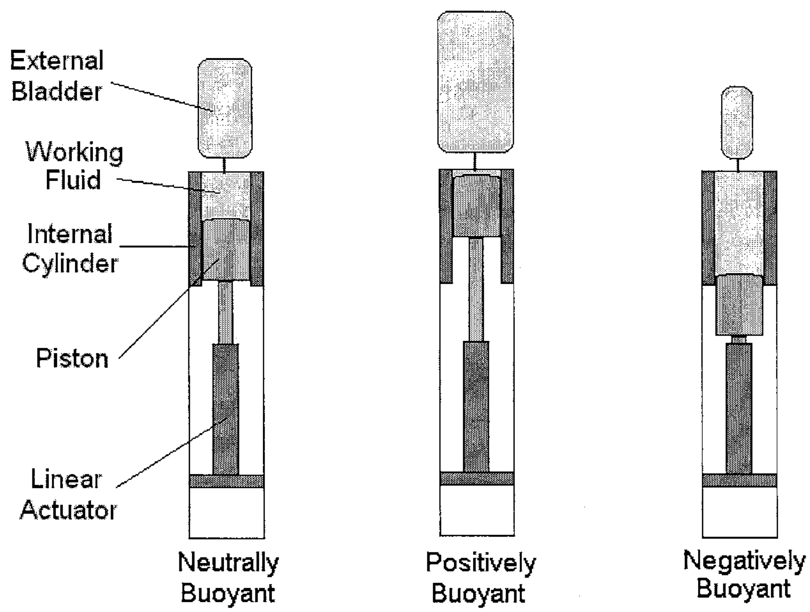


Figure 6 *– Conceptual drawing of Candidate #4.*

3.2.5 Candidate #5

An external submersible pump or internal pump may be used to transfer surrounding fluid from the environment to an internal bladder inside the buoyancy engine. When fluid is transferred into the internal bladder, the weight of the body increases, this will induce a change in buoyant force. The illustration below (Figure 7) shows the system utilizing a submersible pump, but the system may utilize a bi-directional or uni-directional pump located inside the vehicle hull with one fluid line extending outside the vessel.

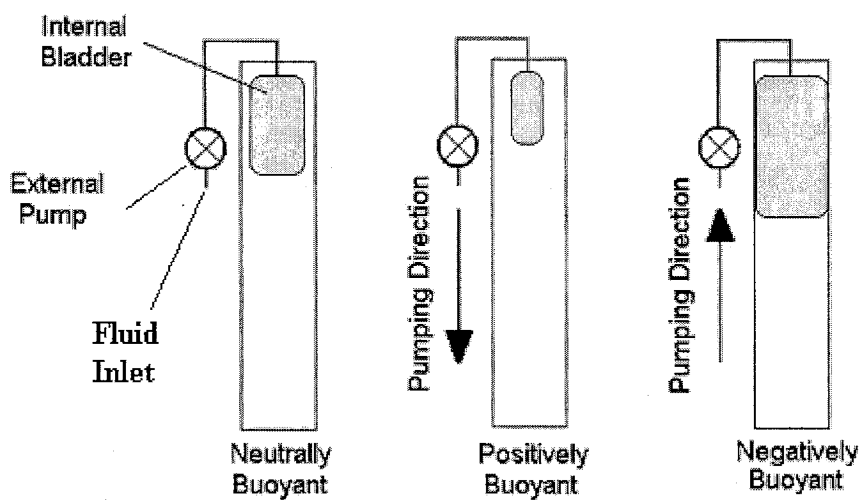


***Figure** 7 –Conceptual drawing of Candidate #5.*

3.3 Mathematical Model of a Buoyancy Engine

Presented in this section will be a mathematical model of the buoyancy engine design. From this model analytical solutions for depth, speed and response time can be calculated which will help in evaluating the candidates.

We will start by modeling the buoyancy engine vehicle as a blunt ended cylinder, in the upright position that will translate in the vertical direction. The coordinate system used will be standard Cartesian xyz with positive z-axis oriented downward. The development of the equation of motion starts by applying Newton's Second Law to the system. The forces acting on the system must sum to the mass of the system multiplied by the acceleration of the system (Figure 8). Summing forces one gets,

$$\sum F = M \cdot \ddot{z}$$

$$W - F_{buoyant} \pm F_{drag} = M \cdot \ddot{z} \tag{1}$$

where,

$F_{buoyant}$ is the buoyant force on the vehicle. [N]
F_{drag} is the drag force on the vehicle. [N]
W is the weight of the vehicle in air. [N]
M is the apparent mass of the vehicle. [kg]
$\ddot{z}$ is the acceleration of the vehicle in z direction. [m/s^2]

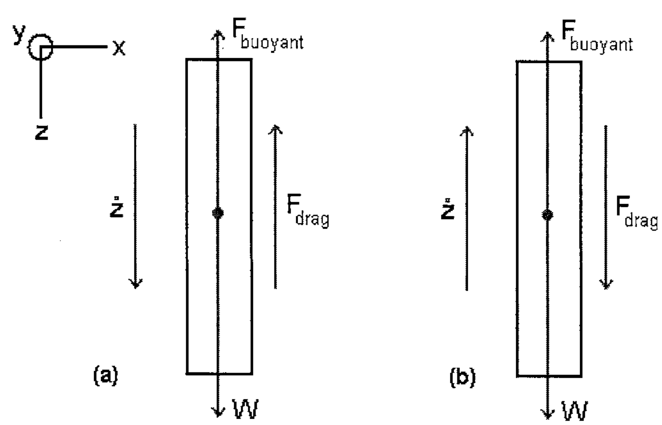


***Figure 8** – Forces acting on buoyancy engine: (a) Forces for motion in positive z direction; (b) Forces for motion in negative z direction.*

The buoyant force is given by,

$$F_{buoyant} = \rho_w g V \tag{2}$$

where,

ρ_w is the density of water. [kg/m^3]

g is the acceleration due to gravity. [m/s^2]

V is the total volume displaced by the vehicle. [m^3]

The drag force is given by,

$$F_{drag} = -\frac{\rho_w A_o c_d}{2} \dot{z} \left| \dot{z} \right| \tag{3}$$

where,

A_o is the cross sectional area of the vehicle. [m^2]

c_d is the drag coefficient for a blunt ended cylinder in axial flow.

$\dot{z}$ is the velocity of the vehicle in the z direction. [m/s]

Note that the direction of the drag force is dependent upon the direction of the vehicle's velocity. If the velocity is positive the drag force direction is negative, likewise if the velocity is negative the drag force direction is positive. The circular cross sectional area of the vehicle is given by,

$$A_o = \frac{\pi d^2}{4}$$

where,

d is the diameter of the vehicle. [m]

The weight of the vehicle is given by,

$$W = mg \tag{4}$$

where,

m is the structural mass of the vehicle. [kg]

Substitution of equations (2), (3) and (4) into equation (1) leads to,

$$M \cdot \ddot{z} = mg - \rho_w g V - \frac{\rho_w A_o c_d}{2} \dot{z} \, | \dot{z} | \tag{5}$$

At this point, a note should be made about the apparent mass M that is given by,

$$M = m + m'$$

where,

m' is the added mass due to the vehicle acceleration. [kg]

For our first analysis of the system we will be solving for the steady state terminal velocity, at this instant the acceleration is zero. Since the acceleration is zero we can take the added mass m' to be zero, thus the apparent mass M simplifies to m and equation (5) becomes,

$$m \cdot \ddot{z} = mg - \rho_w g V - \frac{\rho_w A_o c_d}{2} \dot{z} \, | \dot{z} | \tag{6}$$

Equation (6) is the steady state terminal velocity equation of motion for all the buoyancy engine candidates. By varying the volume V with respect to time, the response of each candidate can be determined. Note that for neutral buoyancy the weight must equal the buoyant force, therefore,

$$\rho_w g V_{vehicle} = mg \tag{7}$$

Also with a change in volume V becomes,

$$V = V_{vehicle} \pm \Delta V \tag{8}$$

where,

$V_{vehicle}$ is the volume of the vehicle at neutral buoyancy. [m^3]

ΔV is a change in the vehicle volume. [m^3]

Substituting equations (7) and (8) into equation (6) one gets the equation of motion for the buoyancy engine with respect to volumetric displacement given by,

$$m \cdot \ddot{z} = \pm \rho_w g \Delta V - \frac{\rho_w A_o c_d}{2} \dot{z} \left| \dot{z} \right| \tag{9}$$

By applying equation (9) to each design candidate, the dynamics of the candidates can be compared and will help in evaluation.

3.3.1 Effect of Volumetric Displacement on Vehicle Motions

Before analyzing each candidate, we must first consider the effect of changes in volume on the analytical model. We must consider what values of terminal velocity are obtained for different changes in volume, at what position the terminal velocities occur and how long of a time interval elapses to reach the terminal velocities.

In this section we analytically solve equation (9) to obtained velocity and position of the vehicle as functions of time. The terminal velocity of the vehicle due to different changes in volume will be calculated and the position and time that the terminal velocities are reached will be approximated. By doing so we will determine if different volumetric changes have any effect on the dynamics of the model. We will start by dividing both sides of equation (9) by m leading to,

$$\ddot{z} = \pm \frac{\rho_w g \Delta V}{m} - \frac{\rho_w A_o c_d}{2m} \dot{z} \left| \dot{z} \right| \tag{11}$$

We will assume downward motion (in the direction of positive z axis) then the absolute value brackets around the velocity in the drag term can be removed. Also, the negative case will be taken for the first term because a negative change in volume will lead to the vehicle descending. With these changes equation (11) becomes,

$$\ddot{z} = -\frac{\rho_w g \Delta V}{m} - \frac{\rho_w A_o c_d}{2m} \dot{z}^2 \tag{12}$$

Next, two new terms will be defined which will simplify the equation for the reminder of the workings. *A* and *B* will be defined as,

$$A = -\frac{\rho_w g \Delta V}{m} \tag{13}$$

$$B = \frac{\rho_w A_o c_d}{2m} \tag{14}$$

Substitution of equations (13) and (14) into equation (12) gives,

$$\ddot{z} = A - B \cdot \dot{z}^2 \tag{15}$$

Note that equation (15) is currently a non-linear ode (ordinary differential equation) and in this format is rather complex to solve. With a substitution one can make the solution more apparent. Using the following relationship,

$$\frac{d\dot{z}}{dt} = \ddot{z}$$

where,

$\frac{d\dot{z}}{dt}$ is the vehicle acceleration in the z direction. [m/s^2]

Equation (15) becomes,

$$\frac{d\dot{z}}{dt} = A - B \cdot \dot{z}^2 \tag{16}$$

Note that the new equation (16) is still a non-linear ode, but now the equation is separable.

Applying separation of variables this leads to,

$$\frac{d\dot{z}}{A - B \cdot \dot{z}^2} = dt \tag{17}$$

Integrating both sides of equation (17) gives,

$$\int \frac{d\dot{z}}{A - B \cdot \dot{z}^2} = \int dt \tag{18}$$

The known hyperbolic tangent integral identity is given by,

$$\int \frac{du}{a - b \cdot u^2} = \frac{1}{\sqrt{a \cdot b}} \tanh^{-1}\left(\sqrt{\frac{b}{a}} \cdot u \right) + c \tag{19}$$

where,

c is a constant of integration.

Using the identity (19) when integrating (18) leads to,

$$\frac{1}{\sqrt{A \cdot B}} \tanh^{-1}\left(\sqrt{\frac{B}{A}} \dot{z} \right) = t + c \tag{20}$$

where,

c is a constant of integration.

The vehicle is starting from rest, therefore when $t = 0$ then $\dot{z} = 0$. Applying this condition to the equation, one finds that the constant of integration is $c = 0$. Thus equation (20) can be written without the constant,

$$\frac{1}{\sqrt{A \cdot B}} \tanh^{-1}\left(\sqrt{\frac{B}{A}} \dot{z} \right) = t \tag{21}$$

Rearrangement of equation (21) will lead to an expression for velocity as a function of time. First multiplying both sides of equation (21) by $\sqrt{A \cdot B}$ gives,

$$\tanh^{-1}\left(\sqrt{\frac{B}{A}}\dot{z}\right) = \sqrt{A \cdot B} \cdot t$$

Using the inverse trig property of hyperbolic tangent this can be written as,

$$\dot{z} = \sqrt{\frac{B}{A}} \cdot \tanh\left(\sqrt{A \cdot B} \cdot t\right)$$

Substituting in the values of A and B from equations (13) and (14) we arrive at the analytical solution for vehicle velocity for a given change in volume as a function of time. This is given as,

$$\dot{z}(t) = \sqrt{\left(\frac{-\rho_w g \Delta V}{m}\right) \Big/ \left(\frac{\rho_w A_o c_d}{2m}\right)} \cdot \tanh\left(\sqrt{\left(\frac{-\rho_w g \Delta V}{m}\right) \cdot \left(\frac{\rho_w A_o c_d}{2m}\right)} \cdot t\right)$$

With manipulation this can be written in simpler form expressed by,

$$\dot{z}(t) = \sqrt{\left(\frac{-2g\Delta V}{A_o c_d}\right)} \cdot \tanh\left(\frac{\rho}{m} \cdot \sqrt{\left(\frac{-gA_o c_d \Delta V}{2}\right)} \cdot t\right) \tag{22}$$

To analytically solve for vehicle position we integrate equation (22) with respect to time. This integration is represented by,

$$\int \dot{z}(t)dt = \int \sqrt{\left(\frac{-2g\Delta V}{A_o c_d}\right)} \cdot \tanh\left(\frac{\rho}{m} \cdot \sqrt{\left(\frac{-gA_o c_d \Delta V}{2}\right)} \cdot t\right) dt \tag{23}$$

The integration can be simplified by defining two new terms D and E as,

$$D = \sqrt{\left(\frac{-2g\Delta V}{A_o c_d}\right)} \tag{24}$$

$$E = \frac{\rho_w}{m}\sqrt{\left(\frac{-gA_o c_d \Delta V}{2}\right)} \tag{25}$$

Substitution of equations (24) and (25) into equation (23) gives,

$$\int \dot{z}(t)\,dt = \int D \cdot \tanh(E \cdot t)\,dt \tag{26}$$

The hyperbolic tangent integration identity is given by,

$$\int \tanh(a \cdot u)\,du = \frac{1}{a}\log(\cosh(a \cdot u)) + c \tag{27}$$

where,

c is a constant of integration.

Using identity (27) and performing the integration on equation (26) leads to,

$$z(t) = \frac{D}{E} \cdot \log(\cosh(E \cdot t)) + c \tag{28}$$

where,

c is a constant of integration.

Substitution of equations (24) and (25) into equation (28) gives,

$$z(t) = \left(\frac{m}{\rho}\right) \cdot \sqrt{\left(\frac{-2g\Delta V}{A_o c_d}\right) \Big/ \left(\frac{-gA_o c_d \Delta V}{2}\right)} \cdot \log\left(\cosh\left(\frac{\rho}{m} \cdot \sqrt{\left(\frac{-gA_o c_d \Delta V}{2}\right)}\right) \cdot t\right) + c$$

Applying the initial condition that $z = 0$ at $t = 0$, we see that the constant of integration $c=0$. With manipulation the equation can be written in a simpler form to give the analytical solution for cylinder position for a given change in volume as a function of time,

$$z(t)=\left(\frac{2m}{\rho A_o c_d}\right)\cdot \log\left(\cosh\left(\frac{\rho}{m}\cdot\sqrt{\left(\frac{-gA_o c_d \Delta V}{2}\right)}\right)\cdot t\right) \tag{29}$$

Now that analytical solutions for vehicle position and velocity for a given change in volume as functions of time have been found, we will use a MATLAB™ script to plot the vehicle's velocity and position for given changes in volume using equations (22) and (29). The script can be found in Appendix A. The changes in volume that will be plotted are 50 cm^3 to 500 cm^3 in increments of 50 cm^3. The results from the MATLAB script are illustrated (Figure 9).

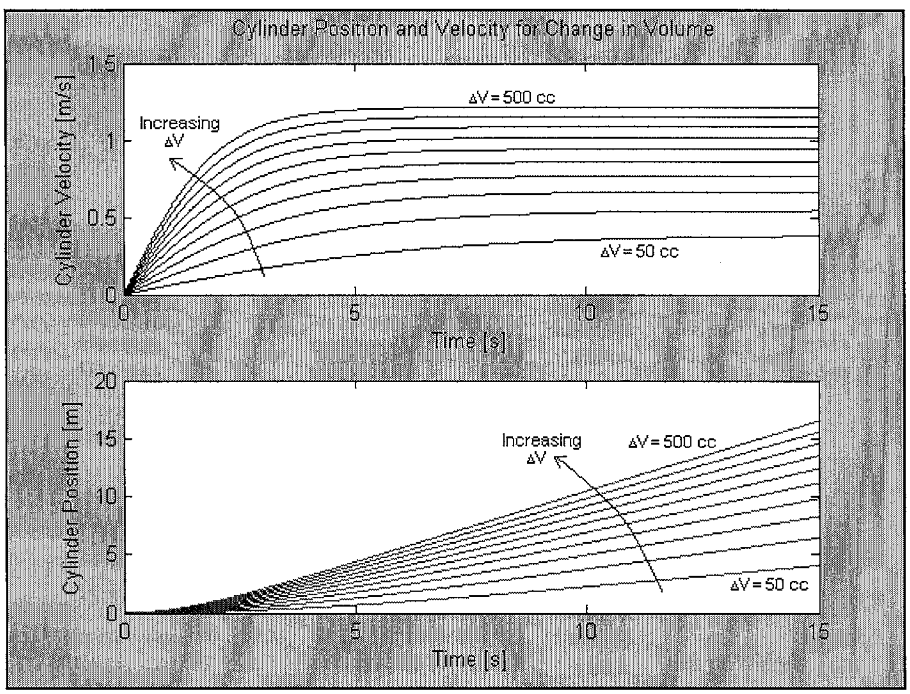


Figure 9 *– Vehicle position and velocity for 10 different volumetric displacements.*

From the plots (Figure 9) it is observed that by increasing ΔV the vehicle's terminal velocity and position increase. It is also shown that the time to reach terminal velocity decreases as ΔV increases.

To calculate analytical values for terminal velocity at each ΔV value we will use equation (12). Knowing that at terminal velocity the vehicle is no longer accelerating, we set $\ddot{z}=0$ and equation (12) becomes,

$$0 = -\frac{\rho_w g \Delta V}{m} - \frac{\rho_w A_o c_d}{2m}\dot{z}^2$$

Rearranging this equation to solve for terminal velocity yields,

$$\dot{z}_{Term} = \sqrt{\frac{-2g\Delta V}{A_o c_d}} \tag{30}$$

where,

$\dot{z}_{Term}$ is the vehicles terminal velocity. [m/s]

Note that equation (30) has a negative sign inside the square root. This will be eliminated since each volumetric change has to be of negative sign in order to decrease the overall volume of the system and cause movement in the direction of the positive z axis. Using equation (30) we can now solve for the analytical terminal velocities at the 10 values of ΔV chosen. The terminal velocity for each ΔV is shown (Table 1).

Change in Volume [cm^3]	Terminal Velocity [m/s]
50	0.38
100	0.54
150	0.67
200	0.77
250	0.86
300	0.94
350	1.02
400	1.09
450	1.15
500	1.21

Table 1 *– Terminal velocities at 10 different volumetric displacements.*

Since the terminal velocities are now calculated, the times at which these velocities occur will be estimated. We will estimate the times at which each terminal velocity is reached using equation (21) given by,

$$\frac{1}{\sqrt{A\cdot B}}\tanh^{-1}\left(\sqrt{\frac{B}{A}}\dot{z}\right)=t \tag{21}$$

where,

$\dot{z}$ is the vehicle velocity in the z direction. [m/s]

Note, that we cannot directly substitute the exact analytical values for terminal velocity into equation (21), because time will end up approaching infinity, as shown on the graphs in Figure 9. Therefore, we will substitute a certain percentage of each terminal velocity, this percentage will be called p. So, as stated above, this will not lead to an exact analytical solution for time, but will rather be used as an approximate value of the time to obtain terminal velocity. Substitution of equations (13), equation (14) and $\dot{z}=p\cdot\dot{z}_{Term}$ into equation (21) leads to the following analytical solution for time as a function of cylinder velocity,

$$t=\left(\frac{m}{\rho_w}\right)\cdot\sqrt{\left(\frac{2}{-gA_o c_d \Delta V}\right)}\cdot\tanh^{-1}\left(\sqrt{\left(\frac{-A_o c_d}{2g\Delta V}\right)}\cdot p\cdot\dot{z}_{Ter\min al}\right) \tag{31}$$

Picking a value of 99% for p as an example, the approximate times at which the vehicle achieves 99% terminal velocity are shown (Table 2).

Volume Change [cm^3]	99% Time [s]
50	17.07
100	12.07
150	9.86
200	8.54
250	7.64
300	6.97
350	6.45
400	6.04
450	5.69
500	5.40

***Table 2** – Effect of ΔV on terminal velocity time.*

Finally the vehicle position at which the terminal velocity is achieved for each ΔV value must be examined to see what effect increasing ΔV has on it. By substituting equation (31), which is the approximation for time at fraction p of the terminal velocity, into equation (29), which is the expression for vehicle position at a given time, we observe equation (29) simplifies to,

$$z = \sqrt{\left(\frac{2m}{\rho_w A_o c_d}\right)} \cdot \log\left(\cosh\left(\tanh^{-1}(p)\right)\right) \tag{32}$$

where,

z is the vehicle position in the z direction. [m]

p is the percentage of vehicle terminal velocity.

Equation (32) shows an important trait of the system that we are interested in. By making the substitution of equation (31) into equation (29), all of the ΔV terms are eliminated, thus the vehicle position at which terminal velocity occurs does not depend whatsoever on the quantity of volume that is displaced by the buoyancy engine. Thus, every possible value we could choose for ΔV will lead to the vehicle achieving terminal velocity at a constant position. Just as an example, using 99% again for p value, and solving for position at various ΔV values, we construct the following table (Table 3).

Volume Change [cm^3]	Terminal Velocity [m/s]	99% Time [s]	Vehicle Position [m]
50	0.38	17.07	4.85
100	0.54	12.07	4.85
150	0.67	9.86	4.85
200	0.77	8.54	4.85
250	0.86	7.64	4.85
300	0.94	6.97	4.85
350	1.02	6.45	4.85
400	1.09	6.04	4.85
450	1.15	5.69	4.85
500	1.21	5.40	4.85

Table 3 *– Summary of the effect of ΔV on vehicle motion.*

3.3.2 Conclusion on the Effect of Volumetric Displacement

From the analytical results for our mathematical model presented in this section we can conclude three important things about the quantity of volume the buoyancy engine should displace. Firstly, as the quantity of volume displaced increases the vehicle's terminal velocity increases, shown graphically (Figure 9) and numerically (Table 1). Secondly, as the quantity of volume displaced increases, the time at which terminal velocity is reached decreases (Table 2). Finally, as the quantity of the volume displaced increases, the vehicle's position at which terminal velocity is achieved remains constant (Table 3). Since the tank for which we are designing this buoyancy engine is 7m deep, and since the terminal velocity is always reached at a constant position (approx 4.85m) there exists a point at which we must change direction to avoid colliding with the bottom of the tank. Therefore any change in volume is acceptable since it does not affect where we must reverse direction. Note, all these conclusions assume an instantaneous step change in volume occurring at $t=0$.

Another important note about these conclusions is that cylinder velocity and position are sensitive to the drag coefficient. This is shown in equations (30) and (32) where we can clearly see that the cylinder velocity and position are both directly proportional to the inverse of the square root of the drag coefficient. The drag coefficient used in our calculations was taken to be 0.82, which was referenced as being the approximate drag coefficient for a blunt ended cylinder in axial flow with a 10:1 length to diameter ratio. By changing the shape of the cylinder the drag coefficient will also change. This will be important if the buoyancy engine is installed into an underwater glider and must be reexamined. The actual drag coefficient, of course will be determined from experiments.

3.4 Components and Performances

The equation of motion developed in Section 3.3 will now be applied to each buoyancy engine candidate. For each candidate, requirements for mechanical components will be defined, and a listing of available parts for each will be listed. Datasheets for all components (pumps, actuators, etc) can be found in Appendix B. The equation of motion will now include mechanical dynamics (i.e. Pump Flow Rate, Linear Actuator Speed etc), thus we will be able to examine what effects displacing the volume at a given rate has on the vehicle's position and velocity. MATLAB scripts are written to plot results for the designs and the scripts are found in Appendix A. We will use a value of 250 cm^3 for volumetric displacement since we can choose any value and this value accommodates all designs. The performance curves will be plotted for each system separately for a simple dive simulation. Then the performances of the candidates will be summarized.

3.4.1 Candidate #1 Components and Performance

We are designing the vehicle to operate up to a depth of 20 metres, therefore the pump used will be subjected to certain pressures. Conventionally it is taken that the pressure increase per 10 metres is 1 bar. Therefore the absolute pressure on the vehicle at 20 metres will be 3 bar, 1 bar due to atmosphere and 2 bar due to depth. Inside the vehicle hull the pressure will be 1 bar, therefore subtracting the outside pressure and inside pressure the differential pressure rating for pumps operating at a 20 metre depth is 2 bar. The bi-directional pumps that have been sourced that satisfy this requirement are listed below (Table 4).

Company	Pump Model	Flow Rate @ Speed	Pressure	Bi - Directional	Cost
Micro Pump	Series 120	3.2 L/min @ 3450 rpm	5.6 Bar Diff	Yes	$1100 US
Micro Pump	Series 120 (GJ)	1.4 to 4.1 L/min @ 4500 rpm	5.6 Bar Diff	Yes	$1100 US
Parker - Oildyne	Miniature 865	2.59 L/min @ 3000 rpm	206 Bar Diff	Yes	$150 US

Table 4 *–Bi-directional pumps with 2 bar differential pressure rating.*

This vehicle candidate also requires an internal and external bladder. The bladders have been chosen to have a capacity of at least 250 cm^3. The bladder that was sourced is listed (Table 5).

Company	Model #	Capacity	Cost
Perma-Type Rubber	PTRTC004	1.23 Litres	$40 US

Table 5– *Inflatable Bladders with at least 250 cm^3capacity.*

At neutral buoyancy, each bladder will contain 250 cm^3 of oil. For the simulation the Micro Pump Series 120 (GJ) pump will be used to transfer oil between the internal and external bladder. To dive the pump will transfer all 250 cm^3 of oil from the external bladder into the internal bladder, the volume decreases as this occurs and the vehicle becomes negatively buoyant. The results from the simulation of this candidate are shown below (Figure 10).

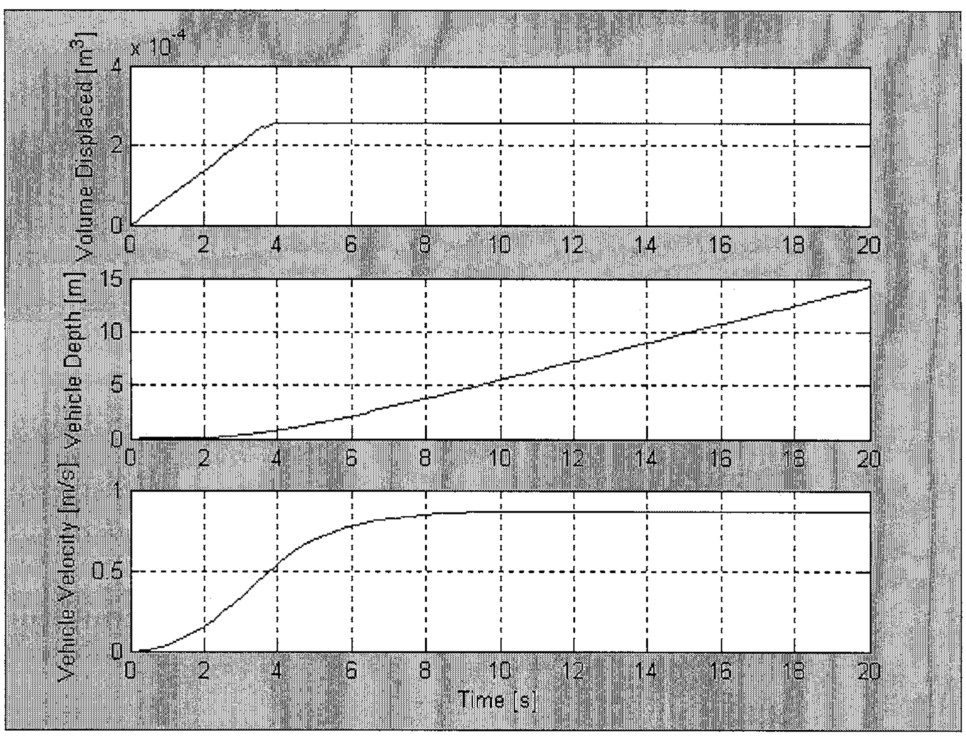


Figure 10 – *Candidate #1 performance curves.*

3.4.2 Candidate #2 Components and Performance

As with the previous design, the pumps for this design must also have a differential pressure rating of at least 2 bar to operate at a depth of 20 metres. Therefore, all uni-directional pumps sourced must satisfy this condition. A table of uni-directional pumps that are satisfactory for this buoyancy engine is below (Table 6).

Company	Pump Model	Flow Rate @ Speed	Pressure	Bi - Directional	Cost
Micro Pump	Series 200	4.0 L/min @ 3450 rpm	8.7 Bar Diff	No	$750 US
Hydro Leduc	PB 36.5	1.8 L/min @ 5000 rpm	350 Bar Diff	No	$1500 US
KNF-Flodos	NF 1.100	1.3 L/min	6 Bar Diff	No	$400 US
Parker - Oildyne	Cartridge Pump	1.65 L/min @ 5000 rpm	206 Bar Diff	No	$150 US

Table 6 *– Uni-directional pumps with 2 bar differential pressure rating.*

This candidate also contains an internal and external bladder. The bladders that have been sourced are identical to those previously listed (Table 5). A flow direction valve is also required to reverse the direction of oil transfer. These valves are typically 12 to 24 volt solenoid operated. A table of solenoid operated flow direction valves usable in this buoyancy engine candidate follows (Table 7).

Company	Valve Model	Type	Cost
HydraForce	SV08 - 40	4-way flow switching	$60 US
Interface Devices	D03-4WPSV	4-way flow switching	$450 US

Table *7 – Four-way flow switching valves.*

As was the case with the bi-directional pump design, at neutral buoyancy this design will have 250 cm^3 of oil in both external and internal bladder. For the simulation Micro Pump Series 200 pump will be used since it has the greatest flow rate. To achieve negative buoyancy 250 cm^3 of oil from the external bladder will be transferred to the internal bladder. The results from the simulation are shown (Figure 11).

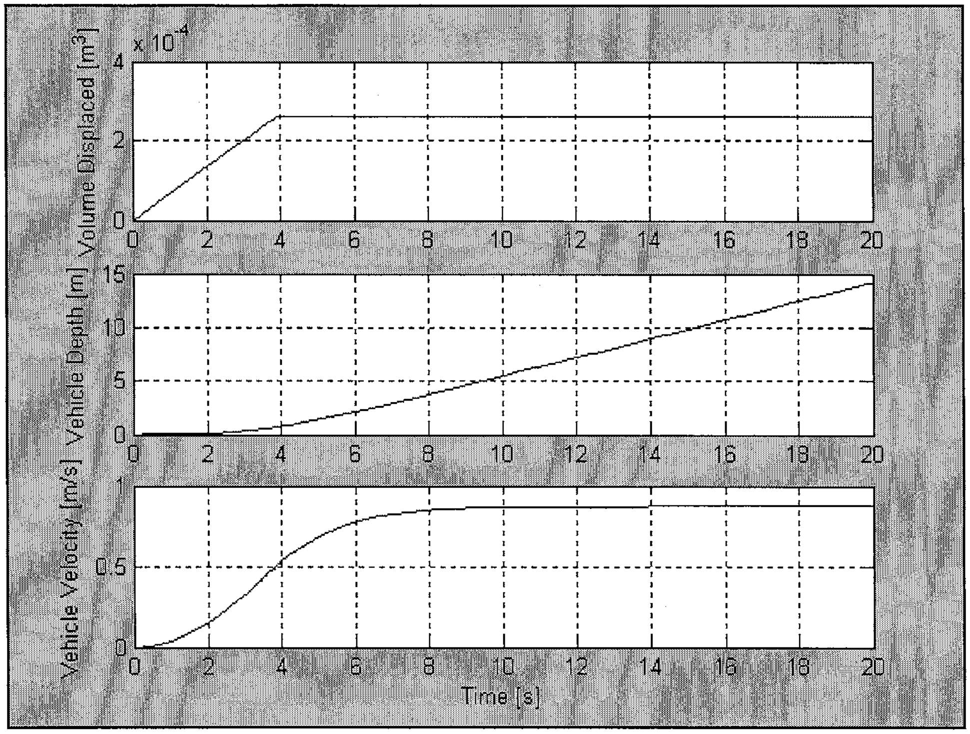


Figure 11 *– Candidate #2 performance curves.*

3.4.3 Candidate #3 Components and Performance

The linear actuators sourced must have enough thrust to push against the diaphragm that is subjected to the pressure felt at 20 metres. Also, the diaphragm itself must not break at this pressure. Another important point to note is to avoid using adhesives to keep the diaphragm in place on the actuator; a 70% vacuum will be created inside the vehicle, which will add to the force acting on the actuator. We will start by determining the pressure on the diaphragm. The pressure felt at a given depth is,

$$P_{depth} = \rho_w g h$$

where,

P_{depth} is the pressure felt at a given depth. [Pa]

ρ_w is the density of water. [kg/m^3]

g is the acceleration due to gravity. [m/s^2]

h is the depth. [m]

Therefore, at 20 metres the pressure felt is,

$$P_{depth} = \left(1000\frac{kg}{m^3}\right)\left(9.81\frac{m}{s^2}\right)(20\,m)$$

$$P_{depth} = 196 \;\; kPa \tag{33}$$

A possible diaphragm sourced that will withstand the given pressure is listed below (Table 8). The diaphragm's half-stroke and effective pressure area was multiplied to get the volume displaced per half-stroke.

Company	Model #	Half-Stroke	E.P.A.	Vol / Half-Stroke	Cost
Dia-Com	D-300-300	6.22 cm	41.0 cm^2	255 cm^3	$550 US

***Table 8** – Diaphragms sourced for Candidate #3.*

The total force felt on the linear actuator at 20 metres will be the combination of force due to depth and force due to vacuum inside the hull. This is given by,

$$F_{Total} = \Delta P \cdot A_{diap} \tag{34}$$

where,

F_{Total} is the total force felt on the actuator. [N]

ΔP is the differential pressure between outside and inside the vehicle. [Pa]

A_{diap} is the effective pressure area of the diaphragm. [m^2]

The differential pressure is given by,

$$\Delta P = P_{depth} + P_{vacuum} \tag{35}$$

where,

P_{vacuum} is the pressure inside the vehicle hull. [Pa]

P_{depth} is the pressure felt at a given depth. [Pa]

The pressure due to 70% vacuum is calculated using,

$$P_{vacuun} = 0.70 \cdot P_{atm}$$

$$P_{vacuum} = 0.70 \cdot (101325\,Pa)$$

$$P_{vacuum} = 70.9\ kPa \tag{36}$$

Solving for the differential pressure, we substitute (36) and (33) into (35) to give,

$$\Delta P = 196\,KPa + 70.9\ kPa$$

$$\Delta P = 267\ kPa$$

Substitution of the differential pressure and the effective pressure area of the diaphragm into equation (34) will give the solution for the required force the linear actuator must supply.

$$F_{Total} = \left(267 \cdot 10^3\ Pa\right) \cdot \left(41.0 \cdot 10^{-4}\ m^2\right)$$

$$F_{Total} = 1095\ N$$

Now that the total thrust that the linear actuator must provide has been determined, several linear actuators were sourced that satisfy the requirements for the design. They are listed (Table 9).

Company	Model #	Type of Motor	Linear Feedback	Linear Speed	Cost
Ultra Motion	Digit	Stepper Motor	Yes	4.57 cm / sec @ 1100 N	$1000 US
IDC	N-Series	24V DC	Yes	1.52 cm / sec @ 1100 N	$1600 US
IDC	N-Series	Stepper Motor	Yes	1.52 cm / sec @ 1100 N	$2000 US
Linak	LA28	24V DC	Yes	2.26 cm / sec @ 1000 N	$300 US
Motion Science	MS400	24V DC or Stepper	Yes	2.54 cm / sec @ 1100 N	$2800 US

Table 9 *– Linear actuators with 1100 N of thrust.*

For the simulation of this candidate, Ultra Motion's Digit linear actuator was used since it is the fastest actuator while at a 1100 N load. To dive, the actuator will retract enough so that the vehicle's volume changes by 250 cm^3. This will lead to the buoyancy engine becoming denser, thus the engine will travel in the positive z axis direction. The results of the simulation of this candidate are shown (Figure 12).

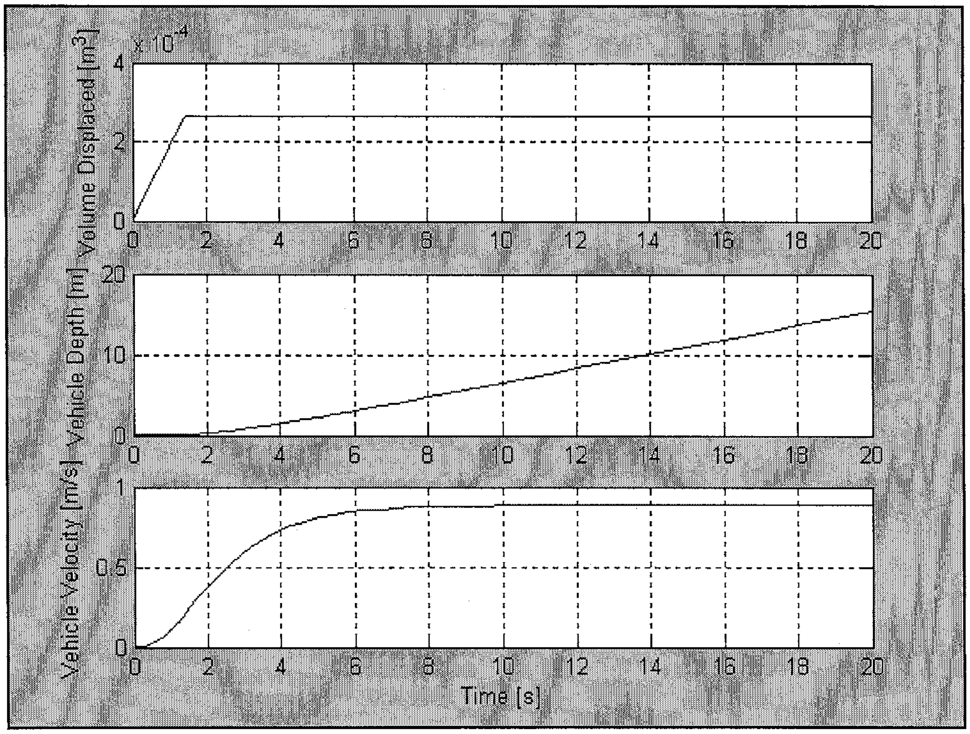


Figure 12 *– Candidate #3 performance curves.*

3.4.4 Candidate #4 Components and Performance

This system uses a linear actuator to push fluid such as oil, between an internal cylinder and an external bladder. The forces on the linear actuator are greatly reduced by using this method, but a much longer stroke will be required to displace the same amount of fluid as opposed to Candidate #3. To start with requirements for the system, we have shown before that the pressure at 20 metre depth is 3 bar and the pressure inside the buoyancy engine is 1 bar, therefore the pressure acting on the face of the piston on top of the linear actuator will be the differential pressure which are these values subtracted, this gives 2 bar. The diameter for the internal cylinder is such that the force acting on the piston will be reduced, thus in choosing a cylinder diameter we get a required thrust given by,

$$F_{Total} = P_{diff} \cdot A_{Piston} \tag{37}$$

where,

F_{Total} is the total force acting on the piston. [N]

P_{diff} is the differential pressure acting on the piston. [Pa]

A_{Piston} is the area of the piston. [m^2]

The area of the piston is given by,

$$A_{Piston} = \frac{d_{Cylinder}{}^2 \cdot \pi}{4} \tag{38}$$

where,

$d_{Cylinder}$ is the diameter of the internal hydraulic cylinder. [m]

Choosing an internal cylinder diameter of 5 cm as an example, and using equations (38) and (37) to calculate the required thrust one gets,

$$F_{Total} = (2\,bar) \cdot \left(100000 \frac{N/m^2}{bar}\right) \cdot \left(\frac{(0.05\,m)^2 \cdot \pi}{4}\right)$$

$$F_{Total} = 405 \;\; N$$

As seen the thrust required by the linear actuator to be sourced is greatly reduced with respect to the linear actuator and diaphragm design, but the drawback is that the half stroke is greatly increased. To achieve a volumetric displacement of 250 cm^3, the half stroke required is given by,

$$L_{Stroke} = \frac{\Delta V}{A_{Piston}} \tag{39}$$

where,

L_{Stroke} is the stroke length. [m]

ΔV is the change in volume. [m^3]

A_{Piston} is the area of the piston. [m^2]

Substitution of equation (38) into equation (39) leads to,

$$L_{Stroke} = \frac{\left(0.000250\,m^3\right)}{\left(\frac{(0.05\,m)^2 \cdot \pi}{4}\right)}$$

$$L_{Stroke} = 12.3 \;\; cm$$

The half stroke calculated leads to an approximate linear actuator length of 25 cm. Available actuators that provide the thrust and stroke requirements were sourced and are listed below (Table 10).

Company	Model #	Type of Motor	Linear Feedback	Linear Speed	Cost
IDC	N-Series	Stepper Motor	Yes	7.62 cm / sec @ 450 N	$2000 US
Motion Science	MS400	24V DC or Stepper	Yes	6.35 cm / sec @ 450 N	$2800 US

***Table 10** – Linear actuators with 25 cm stroke providing 450 N of thrust.*

This buoyancy engine requires an external bladder. The bladder has been chosen to have a capacity of at least 250cc. The bladder that has been sourced is listed (Table 5). In the simulation of this design, the IDC N-Series linear actuator was used since it is the fastest actuator while at a 450 N load. The actuator will contract one full half stroke so that the vehicle's volume changes by 250 cm^3. This will cause the vehicle to move in the positive z direction. The results for the simulation of this candidate are shown below (Figure 13).

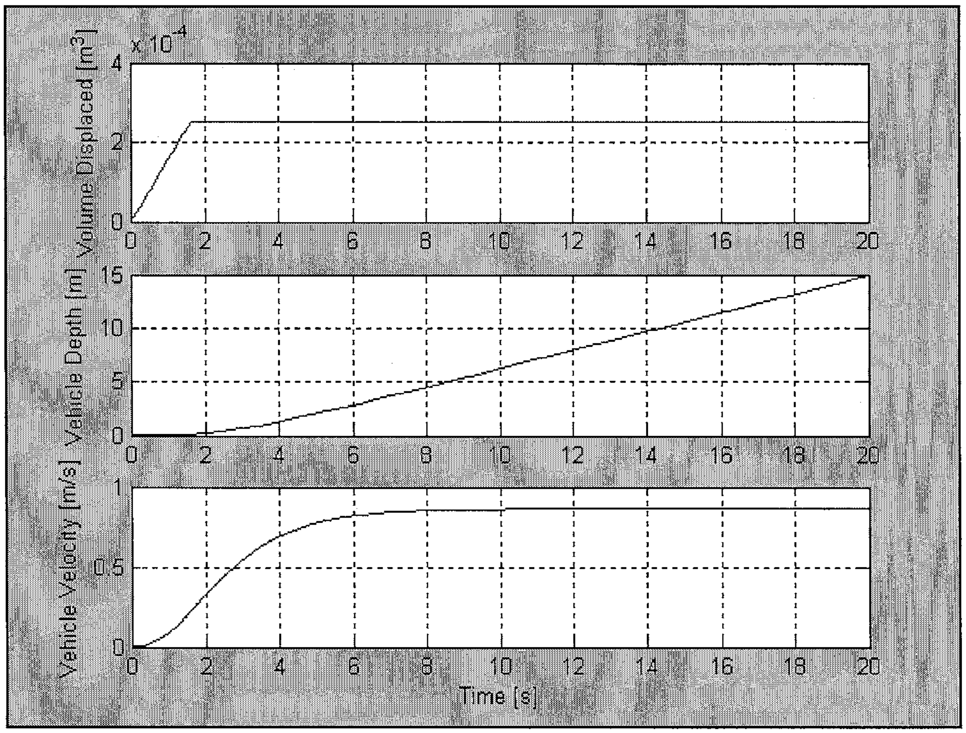


***Figure 13** – Candidate #4 performance curves.*

3.4.5 Candidate #5 Components and Performance

The pump required for this design must have a differential pressure rating of 2 bar like other candidates with pumps. The pump can either be bi-directional or uni-directional; in the uni-directional case a directional flow valve must be utilized to pump fluid both ways. The pump may also be submersible in which case it can be located outside the pressure hull. The pumps that have been sourced for this design are given (Table 11).

Company	Pump Model	Flow Rate @ Speed	Pressure	Bi - Directional	Cost
Micro Pump	Series 120	3.2 L/min @ 3450 rpm	5.6 Bar Diff	Yes	$1100 US
Micro Pump	Series 120 (GJ)	1.4 - 4.095 L/min @ 4500 rpm	5.6 Bar Diff	Yes	$1100 US
Parker - Oildyne	Miniature 865	2.59 L/min @ 3000 rpm	206 Bar Diff	Yes	$150 US
Micro Pump	Series 200	4.0 L/min @ 3450 rpm	8.7 Bar Diff	No	$750 US
Hydro Leduc	PB 36.5	1.8 L/min @ 5000 rpm	350 Bar Diff	No	$1500 US
KNF-Flodos	NF 1.100	1.3 L/min	6 Bar Diff	No	$400 US
Parker - Oildyne	Cartridge Pump	1.65 L/min @ 5000 rpm	206 Bar Diff	No	$150 US

***Table 11**– Pumps with 2 bar pressure rating.*

This design candidate also requires an internal bladder with a capacity of at least 250 cm^3. The bladders that have been sourced have been listed previously (Table 5). When conducting the dive simulation for this candidate, the Micro Pump Series 120 (GJ) pump will be used since it is the fastest pump out of all that were sourced. This is a bi-directional pump unit and will not require the use of a flow direction valve. Also, this unit is not submersible so the pump will be assumed to be located inside the pressure hull with one of the fluid lines extending outside the pressure hull into the surrounding fluid. The system will initialize with the internal bladder containing 250 cm^3 of water (neutral buoyancy). The pump will transfer 250 cm^3 of water surrounding the vehicle into the vehicle's internal bladder. This will cause an increase in the vehicle's weight and it will dive. The simulation produced the following results for vehicle position and velocity (Figure 14).

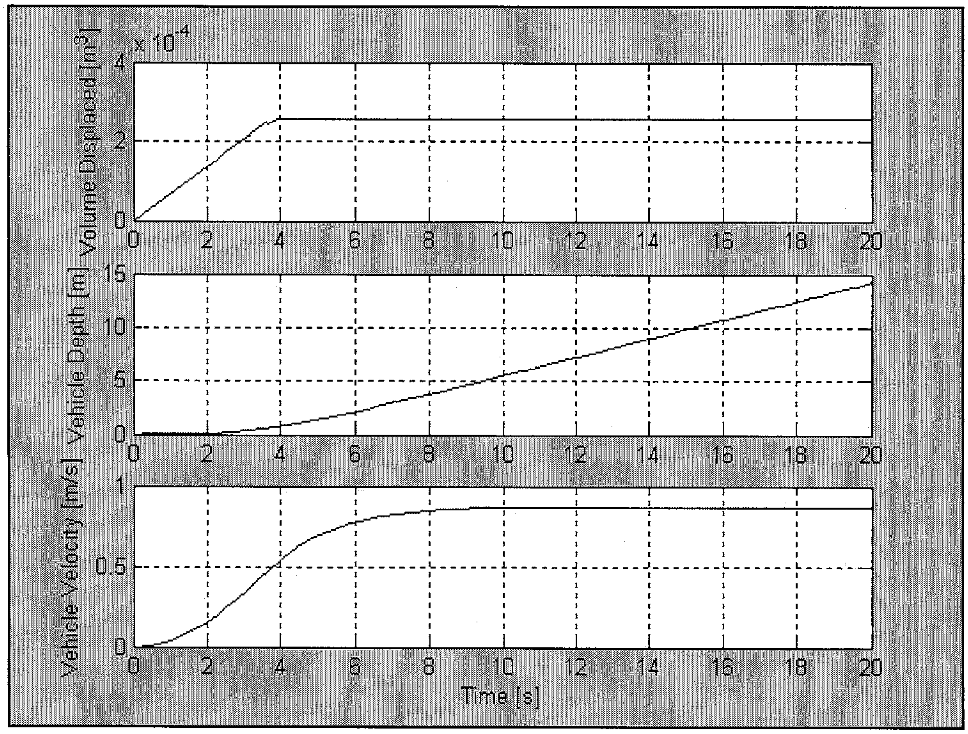


***Figure 14** – Candidate #5 performance curves.*

3.5 Performance Comparison

In this section the velocity performance curves of each candidate are normalized and superimposed so that we can compare each candidate to one another. The curves of velocity vs. position for each candidate will also be superimposed. All the plots in this section will be generated using MATLAB; all the programs written can be found in Appendix A. To normalize each velocity plot, we will divide the velocity vs. time plot of each candidate by that candidate's terminal velocity, thus each plot will converge to unity and the five curves are then comparable. When superimposing all velocity plots from the design candidates we construct the following graph (Figure 15).

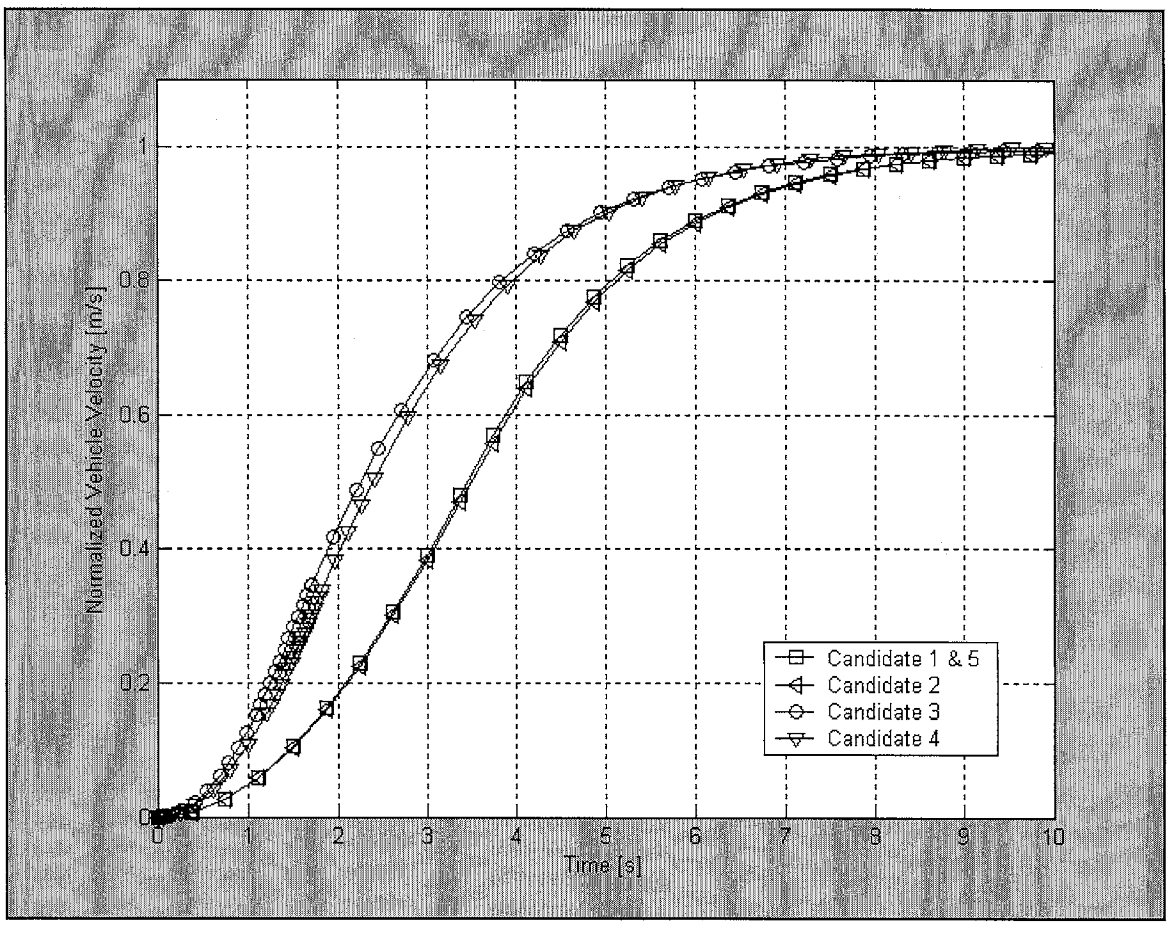


Figure 15 *– Candidate normalized vehicle velocity vs. time curves.*

Similarly when plotting velocity vs. position for each buoyancy engine design we will normalize the velocities of each engine first. Using another program we construct the following graph (Figure 16).

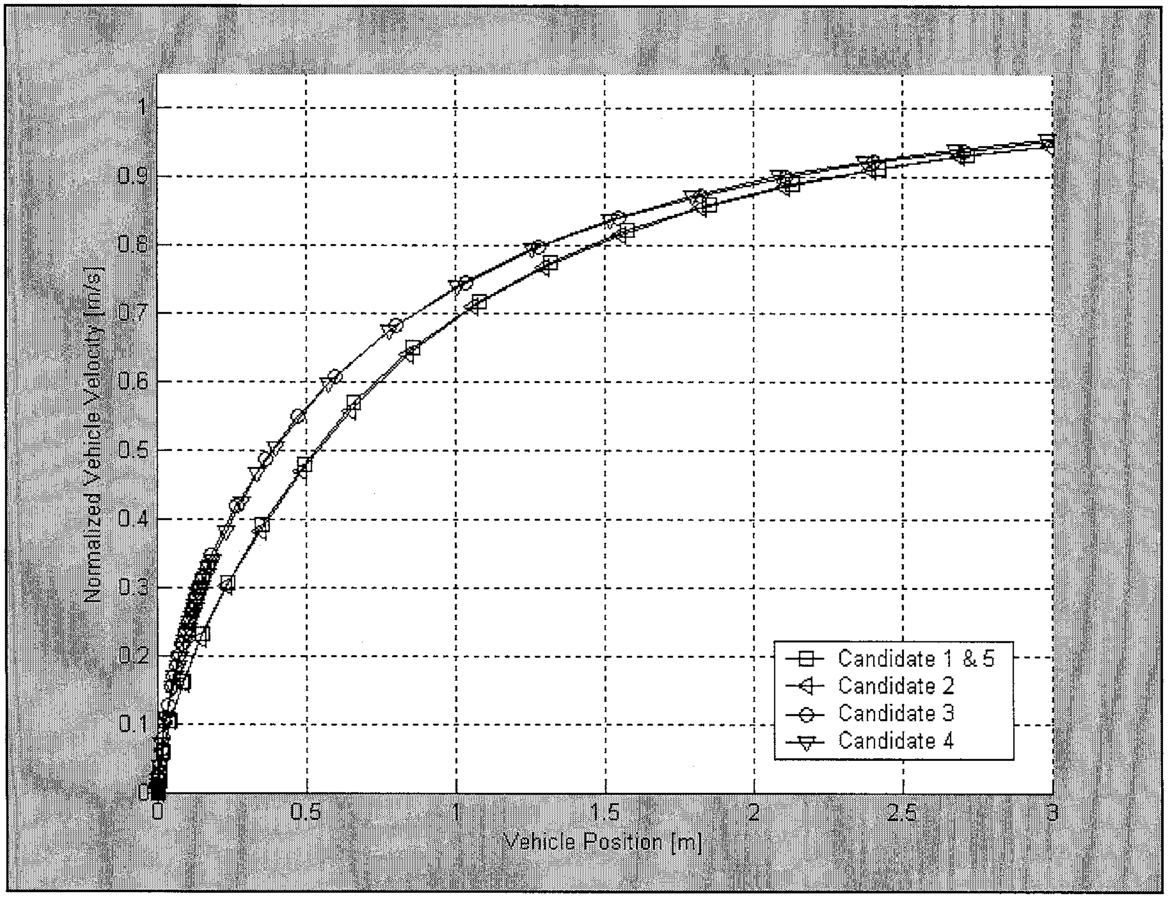


Figure 16 *– Candidate normalized vehicle velocities vs. position curves.*

From these plots (Figures 15 and 16) we clearly see that the design candidates utilizing pumps to displace the volume (Candidates #1,2,5) are slower then the design candidates that use linear actuators to displace the volume (Candidates #3,4). This is as expected, since from examination of each candidate's performance curves we can clearly see that the linear actuator designs displace the volume faster so we conclude that they would change vehicle motions faster. The design candidate that changes the dynamics of the system the fastest is Candidate #3.

3.6 Cost Comparison

We will now present a simple cost comparison of the design candidates. The problem with a full cost comparison is that each system will require significant machining, along with other components and to get an exact cost analysis would require the design of each candidate. Therefore to present an approximate cost comparison where the candidates can be compared relative to one another we must assume that machining costs and other additional costs are constant across all design candidates. We will also conduct the comparison using components utilized in the dive simulation of each candidate since those used gave best performance, which is ultimately what we are interested in achieving. With these assumptions in place we calculate the approximate costs of each candidate by summing the costs of the major components that have been introduced previously (Tables 4 to 11). Summing the components we construct a table that lists the approximate cost of each candidate (Table 12).

Candidate Number	Relative Cost
1	$1,180
2	$1,280
3	$1,550
4	$2,040
5	$1,140

***Table 12** – Cost Comparison of the design candidates.*

From the presented costs we can conclude that the most expensive design candidate will be candidate 4, while the least expensive would be candidate 5. The other three candidate's costs fall roughly in the middle of these two extremes. Note that these are not final costs because addition materials, electronics, machining and assembly costs have not been factored into these values but we are assuming these additional costs are constant across all designs for this comparison.

3.7 Energy Comparison

The design candidate's approximate energy consumption will now be compared. To do this we will compare how many approximate cycles each design candidate can produce from a 24 volt / 6 Ah battery. A cycle is considered the displacement of 250 cm^3 of volume, which was the amount used in the simulation of each candidate. First, the time each system takes to displace the volume will be determined using,

$$T_{\Delta V} = \frac{2.5 \cdot 10^{-4}\, m^3}{C} \tag{40}$$

where,

$T_{\Delta V}$ is the time taken to displace the volume. [s]
C is the volumetric flow rate of the candidate. [m^3/s]

Now plugging values into equation (40) we can construct a table listing the analytical times that each candidate would required to displace 250 cm^3 (Table 13).

Candidate Number	Volumetric Flow Rate [m^3/s]	$T_{\Delta V}$ [s]
1	6.83 (10^{-3})	3.66
2	6.66 (10^{-3})	3.75
3	1.88 (10^{-3})	1.33
4	1.47 (10^{-3})	1.70
5	6.83 (10^{-3})	3.66

***Table 13** – Times for candidates to displace 250 cm^3.*

Note that these values can also be interpreted from figures 10 to 14 by examining the Volume Displaced vs. Time plots. But, by calculating the times analytically any human error in measurement due to reading plots inaccurately is eliminated. Next we must examine the amperage each design draws during each cycle. By multiplying the amperage drawn during a cycle, by the cycle time and 24 volts we get the Watt-seconds of power consumption during the cycle. Then, dividing the overall Watt-seconds of power the battery can supply by the Watt-seconds drawn during one cycle we calculate the number of cycles achieved by the design candidate. Doing this for each design candidate a table was constructed showing the number of cycles each design candidate would achieve (Table 14).

Candidate Number	Volts / Amps	Cycles
1	24/1.90	3106
2	24/2.75	2094
3	24/3	5413
4	24/3	4235
5	24/1.90	3106

***Table 14** – Number of cycles of operation for design candidates.*

Examining the table above (Table 14), we conclude that candidate #3 has the best energy consumption rate because it can achieve 5413 cycles on a 24 volt / 6 Ah battery. Candidate #2 ended up being the worst in terms of power consumption due to only being able to achieve 2094 cycles before depleting the battery. This information will help in choosing the most suitable candidate since it is desirable to achieve the most cycles possible.

3.8 Conclusion of Comparison

Since we are ultimately interested in the candidate that provides the best performance Candidate #3 would be more suitable for our purpose. This particular design candidate falls in the mid ranges of design costs being roughly $1550 US dollars for the major components. Candidate #3's performance gain over Candidate #5 (being the least expensive design) far outweighs the added cost (approx ~$500). In terms of energy consumption, Candidate #3 can achieved more cycles of operation then any of the other design candidate's (achieving 5413 cycles from a 24v/6Ah battery). Therefore due to these affirmative traits Candidate #3 will be selected as the best candidate and will be designed. The following section will detail the mechanical layout of the chosen buoyancy engine.

4.0 Design Details

This section will detail the design layout of Candidate #3. An illustration of this design is shown in the picture below (Figure 17). The major components have all been labeled for clarity. The design drawings and assembly drawings for all pieces can be found in Appendix C, where they may be referenced for dimensions. The linear actuator that will be used is the UltraMotion Digit linear actuator with the NEMA 23 stepper motor and 8 inch long stroke. A Dia-Com Type D-300-300 rolling rubber diaphragm will be selected as the diaphragm unit. The End Cap, Electronics Tray Support Rails, Mounting Plate Supports, Diaphragm Mount, and Diaphragm Fastener components will all be constructed out of PVC material. The Electronics Tray and Actuator Mounting Plate will be constructed from Aluminum. The piston will be made from stainless steel.

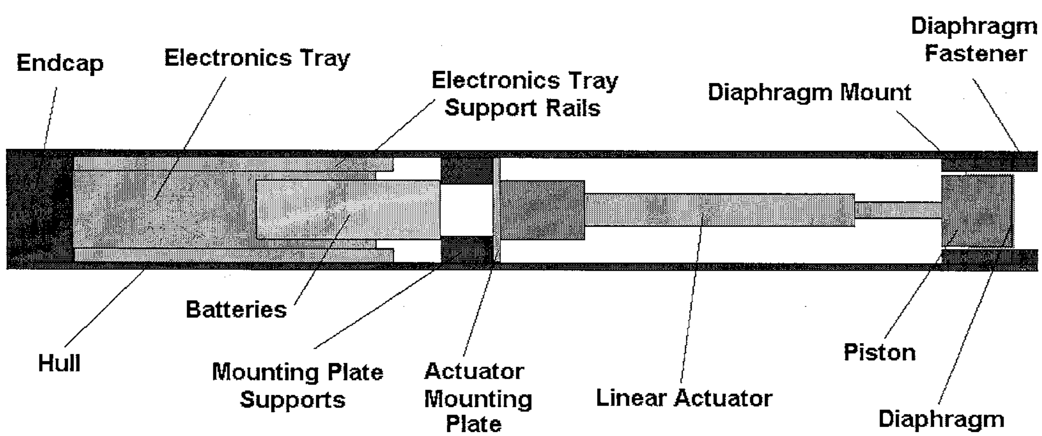


***Figure 17** –Layout of the buoyancy engine.*

The structural assembly would involve several steps, which are detailed in Appendix C. First the linear actuator is mounted onto the mounting plate then the four mounting plate supports are bolted onto the mounting plate. The piston is then bolted onto the top of the linear actuator. Next, these assembled parts are inserted into the end cap side of the engine and the four mounting supports are glued to the interior hull wall. Following this the diaphragm mount is glued into place on the interior of the hull as well. The electronics tray is inserted into the tray rails and then this unit is inserted into the hull and the rails are glued to the interior. An o-ring is placed on the end cap, which then screws into the hull. Finally the diaphragm is fitted into the grooves on the diaphragm mount and over the piston, a Teflon ring is placed on top of the diaphragm and the diaphragm fastener is then screwed into the hull thus clamping the diaphragm. A hand tool is constructed out of aluminum to screw the diaphragm fastener down. When batteries or electronics need to be replaced the end cap can be unscrewed and the electronics tray and batteries can be removed easily.

Upon beginning the mechanical design of the buoyancy engine, an important point arose about the positions of the center of gravity relative to the center of buoyancy. For testing the buoyancy engine's vertical translation, the center of gravity of the vehicle must be below the center of buoyancy of the vehicle at all times or the vehicle will flip over due to the moment created by an offset of the two centers. Therefore the situation illustrated above (Figure 17) displays the situation where the batteries have been placed near the middle of the buoyancy engine on purpose. Since the batteries and the actuator contain a large portion of the buoyancy engine mass, the center of gravity will be shifted toward the actuator side of the engine. The engine will translate vertically with the piston side pointing downward, thus the center of gravity will be below the center of buoyancy upon positioning the batteries near the middle. Using MATLAB a plot of the locations of the center of gravity and the center of buoyancy relative to actuator location was constructed (Figure 18). The center of gravity and center of buoyancy are measured from reference A, while the actuator location is measured from reference B (Figure 18). The plot shows that for the full range of motion that the actuator is capable of, the center of gravity is always located below the center of buoyancy thus the buoyancy engine will remain in its upright vertical position during its vertical translation. Note that once the buoyancy engine is turned with its longitudinal axis oriented horizontally there will exists an offset in the horizontal plane between the center of buoyancy and center of gravity.

If this buoyancy engine is to be utilized for an underwater glider this offset in the center of gravity and center of buoyancy might be undesirable since the moment created could cause the glider to tilt. Therefore, once this engine is used in an underwater glider the centers will have to be adjusted. There are several ways this can easily be achieved. One method to fix this would simply be to move the batteries into the rear section of the buoyancy engine since they contain a fair amount of the mass that occupies the mid section. Also the piston could be made lighter, out of aluminum instead of steel, and the extra mass gained from this could be moved to the rear of the vehicle by adding some weights near the electronics tray.

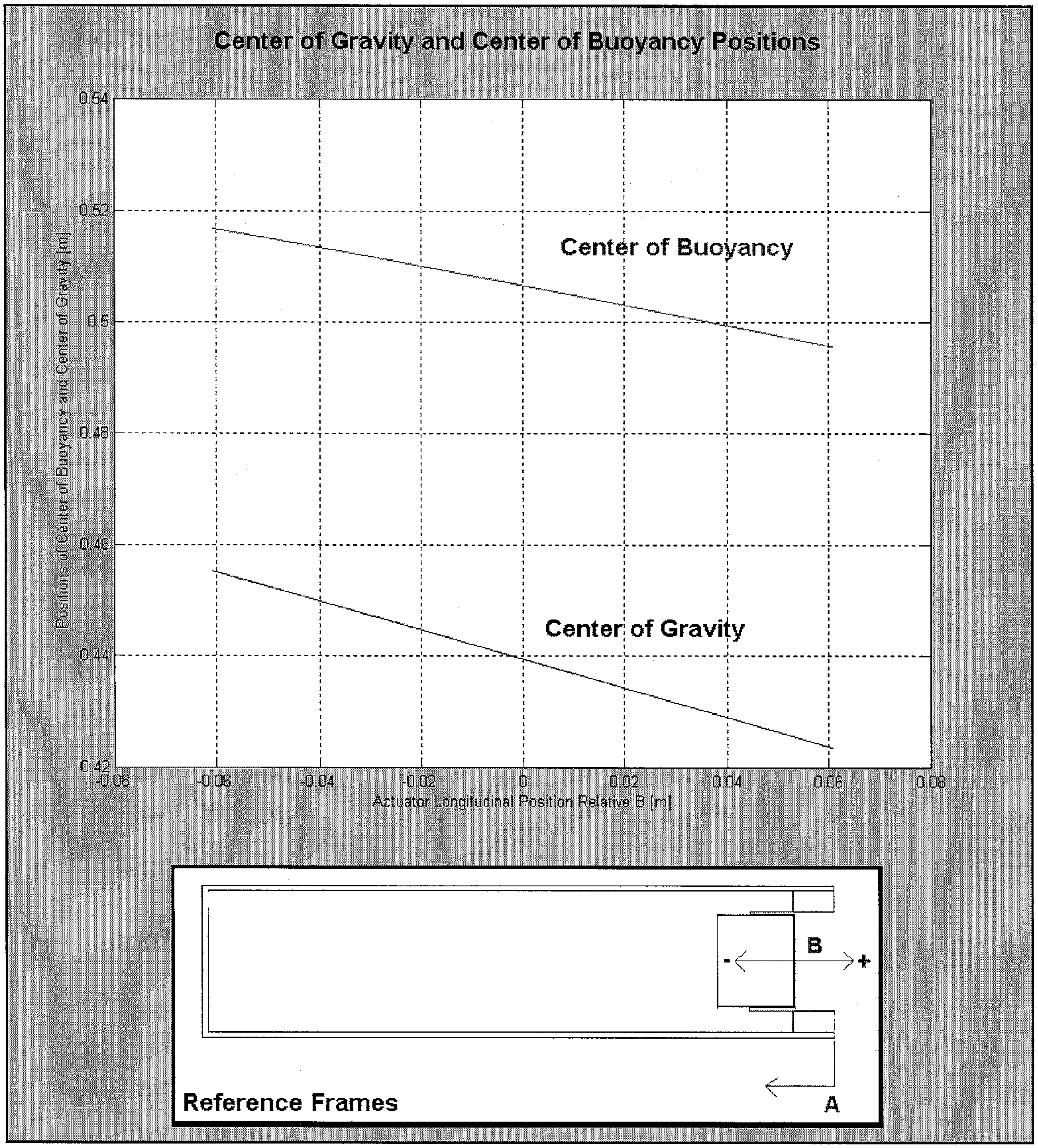


***Figure 18** – Center of gravity and center of buoyancy location.*

5.0 Conclusions and Future Work

In this report five designs that were possible candidates for a buoyancy engine for an underwater glider were examined. A mathematical model was developed to see the differences in the dynamics between the five candidates. Using the mathematical model vehicle velocity, position and displaced volume were all plotted against time. Normalizing these plots and superimposing them lead to us to evaluate the dynamics of each of the candidates relative to one another. Our results showed that the system that had the fastest response time was Candidate #3, utilizing a linear actuator and rolling diaphragm to create a change in volume. Following the performance analysis the costs and power consumption of the five candidates were also examined. Upon reviewing the performances, costs and power consumptions we determined that Candidate #3 was indeed the most suitable design for our purposes. The report then detailed the structural layout of Candidate #3 and a note about center of buoyancy relative center of gravity was discussed.

Future work would include the construction of a prototype buoyancy engine. This would include the purchasing of components and materials required, machining several parts, and assembly of the unit. A pressure sensor would have to be purchased to provide feedback for the depth of the buoyancy engine. One sensor that is suitable for this was sourced and is located in Appendix B. The best location for this pressure sensor would be in the end cap component. This would require a hole to be drilled and tapped in the end cap so that the pressure sensor could be installed. The prototype buoyancy engine could then be used to verify the analytical solutions attained throughout this report and from them other conclusions can be made. Once the buoyancy engine is installed into an underwater glider the center of gravity and center of buoyancy issues discussed at the end of Section 4.0 will have to be examined again.

APPENDIX A: MATLAB PROGRAMS

```
%-------------------------------------------------
% Name:      Plots for Velocity and Position
%            for 10 different changes in Volume
%
% Author:  Nicholas Janes
% Date:    March 6 2004
% NRC-IOT
%----( Description )------------------------------
% This code generates Figure 9 in Section 3.3.1
% Plots of Velocity and Position for 10 changes in
% volume are plotted from 50 cc to 500 cc in
% increments of 50 cc.
%-------------------------------------------------
function volposV

% equation constants
inch_2_m = 0.0254;          % conversion factor [m/inch]
g = 9.81;                   % gravity [m/s^2]
rho = 1000;                 % density of water [kg/m^3]
D = 4 * inch_2_m;           % diameter of cylinder [m]
L = 40 * inch_2_m;          % length of cylinder [m]
Ao = (D^2)*pi/4;            % cross sectional area of cylinder [m^2]
V = Ao * L;                 % volume of cylinder [m^3]
M = V * rho;                % mass of cylinder [kg]
Cd = 0.82;                  % drag coefficient for L:D of 10:1

% time vector
t = [0:0.01:15];

% loop through various changes in volume
% plotting cylinder position and velocity
% for each case.
for i = 1:10

   DeltaV = -0.00005*i;   % change in volume [m^3]

   % Note: DeltaV must be a negative value
   %      because the volume of the buoyancy engine
   %      must decrease for the engine to travel
   %      in the positive z direction.

   % cylinder position and velocity equations
   position = (2*M/(rho*Ao*Cd)) * log(cosh(t*(rho/M)*(-g*Ao*Cd*DeltaV/2)^(0.5)));
```

```
velocity = tanh(t*(rho/M)*(-g*Ao*Cd*DeltaV/2)^(0.5)) * (2*g*DeltaV/(Ao*Cd))^(0.5);

   % plotting velocity
   figure(1);
   subplot (2,1,1);
   plot(t,velocity,'b');
   hold on
   ylabel('Cylinder Velocity [m/s]');
   xlabel('Time [s]');

   % plotting position
   subplot (2,1,2);
   plot(t,position,'r');
   hold on
   ylabel('Cylinder Position [m]');
   xlabel('Time [s]');

end
```

```
%--------------------------------------------------------
% Buoyancy Engine Simulator
% Candidate 1 - Bi-Directional Pump Buoyancy Engine
% Nick Janes
% Feb 25 2004
% NRC-IOT
%---(Description)--------------------------------------
% This is the code used for the simulation in
% Section 3.4.1. The command line input is
% located at the end of the script.
%--------------------------------------------------------
function x_out = candidate_1(t,x)

% constants ---------------------------------------------
inch_2_m = 0.0254;     % conversion factor from inch to meters [m/inch]
g = 9.81;              % acceleration due to gravity [m/s^2]
rho = 1000;            % density of water [kg/m^3]
D = 4 * inch_2_m;      % diameter of the cylinder [m]
L = 40 * inch_2_m;     % length of the cylinder [m]
Ao = (D^2)*pi/4;       % cross sectional area of cylinder [m^2]
V = Ao * L;            % volume of the cylinder [m^3]
M = V * rho;           % mass of the cylinder (taking added mass to be zero) [kg]
Cd = 0.82;             % coefficient of drag for cylinder in axial flow

% pump constants ----------------------------------------
pump_speed = 6.825 * 10^(-5);  % flow speed [m^3/s]
pump_max_speed = pump_speed;   % max pump speed
pump_min_speed = -pump_speed;  % min pump speed

% bladder constants -------------------------------------
bladder_max = 0.000250;     % bladder upper limit [m^3]
bladder_min = -0.000250;    % bladder lower limit [m^3]

x1 = x(1);  % reset bladder contents [m^3]
x2 = x(2);  % reset vehicle depth [m]
x3 = x(3);  % reset vehicle velocity [m/s]

% limit overfilling of the bladders ----------------------
if (x1 >= 0 & x1 <= bladder_max)
   x1_dot = pump_max_speed;
elseif (x1 < 0 & x1 >= bladder_min)
   x1_dot = pump_min_speed;
else
   x1_dot = 0;
end
```

```
% make sure not to overfill past 250 cm^3 -----------------
if (x1 > bladder_max)
   x1 = bladder_max;
end
if (x1 < bladder_min)
   x1 = bladder_min;
end

x2_dot = x3;
x3_dot = -rho * g * x1 / M - 1/2 * rho * Ao / M * Cd * x3 * abs(x3);        % equation 9
x_out = [x1_dot;x2_dot;x3_dot];
```

Comand Line Input to run the program:

```
>> [t,x]=ode45('candidate_1',[0 20],[0 0 0]);

>> figure(1)

>> subplot(311)
>> plot(t, x(:,1));
>> grid
>> ylabel ('Volume Displaced [m^3]');

>> subplot(312)
>> plot(t,-x(:,2));
>> grid
>> ylabel ('Vehicle Depth [m]');

>> subplot(313)
>> plot(t,-x(:,3));
>> grid
>> ylabel ('Vehicle Velocity [m/s]');

>> xlabel ('Time [s]');
```

```
%-------------------------------------------------------
% Buoyancy Engine Simulator
% Candidate 2 - Uni-Directional Pump Buoyancy Engine
% Nick Janes
% Feb 25 2004
% NRC-IOT
%---(Description)--------------------------------------
% This is the code used for the simulation in
% Section 3.4.2. The command line input is
% located at the end of the script.
%-------------------------------------------------------
function x_out = candidate_2(t,x)

% constants -----------------------------------------------
inch_2_m = 0.0254;   % conversion factor from inch to meters [m/inch]
g = 9.81;            % acceleration due to gravity [m/s^2]
rho = 1000;          % density of water [kg/m^3]
D = 4 * inch_2_m;    % diameter of the cylinder [m]
L = 40 * inch_2_m;   % length of the cylinder [m]
Ao = (D^2)*pi/4;     % cross sectional area of cylinder [m^2]
V = Ao * L;          % volume of the cylinder [m^3]
M = V * rho;         % mass of the cylinder [kg]
Cd = 0.82;           % coefficient of drag for cylinder in axial flow

% pump constants -----------------------------------------
pump_speed = 6.6666667 * 10^(-5);      % flow rate [m^3/s]
pump_max_speed = pump_speed;           % max pump speed [m^3/s]
pump_min_speed = -pump_speed;          % min pump speed [m^3/s]

% bladder constants --------------------------------------
bladder_max = 0.000250;          % bladder upper limit [m^3]
bladder_min = -0.000250;         % bladder lower limit [m^3]

x1 = x(1);           % reset bladder contents [m^3]
x2 = x(2);           % reset vehicle depth [m]
x3 = x(3);           % reset vehicle velocity [m/s]

% limit overfilling of the bladders ----------------------
if (x1 >= 0 & x1 <= bladder_max)
   x1_dot = pump_max_speed;
elseif (x1 < 0 & x1 >= bladder_min)
   x1_dot = pump_min_speed;
else
   x1_dot = 0;
end
```

```
% make sure not to overfill past 250 cm^3
if (x1 > bladder_max)
   x1 = bladder_max;
end
if (x1 < bladder_min)
   x1 = bladder_min;
end

x2_dot = x3;
x3_dot = -rho * g * x1 / M - 1/2 * rho * Ao / M * Cd * x3 * abs(x3);        % equation 9
x_out = [x1_dot;x2_dot;x3_dot];
```

Comand Line Input to run the program:

```
>> [t,x]=ode45('candidate_2',[0 20],[0 0 0]);

>> figure(1)

>> subplot(311)
>> plot(t, x(:,1));
>> grid
>> ylabel ('Volume Displaced [m^3]');

>> subplot(312)
>> plot(t,-x(:,2));
>> grid
>> ylabel ('Vehicle Depth [m]');

>> subplot(313)
>> plot(t,-x(:,3));
>> grid
>> ylabel ('Vehicle Velocity [m/s]');

>> xlabel ('Time [s]');
```

```
%------------------------------------------------------------
% Buoyancy Engine Simulator
% Candidate 3 - Linear Actuator and Rolling Diaphragm
% Nick Janes
% Feb 25 2004
% NRC-IOT
%---(Description)--------------------------------------------
% This is the code used for the simulation in
% Section 3.4.3. The command line input is
% located at the end of the script.
%------------------------------------------------------------
function x_out = candidate_3(t,x)

% constants ------------------------------------------------
inch_2_m = 0.0254;          % conversion factor from inch to meters [m/inch]
g = 9.81;                   % acceleration due to gravity [m/s^2]
rho = 1000;                 % density of water [kg/m^3]
D = 4 * inch_2_m;           % diameter of the cylinder [m]
L = 40 * inch_2_m;          % length of the cylinder [m]
Ao = (D^2)*pi/4;            % cross sectional area of cylinder [m^2]
V = Ao * L;                 % volume of the cylinder [m^3]
M = V * rho;                 % mass of the cylinder (taking added mass to be zero) [kg]
Ad = 6.35 * inch_2_m^2;     % effective pressure area of the diaphragm [m^2]
Cd = 0.82;                  % coefficient of drag for cylinder in axial flow

% actuator parameters --------------------------------------
actuator_halfstroke = 2.4025 * inch_2_m; % halfstroke to displace 250 cm^3 length [m]
actuator_speed = 1.8 * inch_2_m;          % actuator speed [m/s]

% actuator limits ------------------------------------------
x1d_max = actuator_speed;           % max actuator speed [m/s]
x1d_min = -actuator_speed;          % min actuator speed [m/s]
x_min = -actuator_halfstroke;       % negative halfstroke [m]
x_max = actuator_halfstroke;        % positive halfstroke [m]

x1 = x(1);          % actuator stroke [m]
x2 = x(2);          % vehicle position [m]
x3 = x(3);          % vehicle velocity [m/s]

% we have to limit the extension of the actuator
if (x1 >= 0 & x1 <= x_max)
   x1_dot = x1d_max;
elseif (x1 < 0 & x1 >= x_min)
   x1_dot = x1d_min;
else
   x1_dot = 0;
```

```
end

% make sure not to displace past 250 cm^3 either way
if (x1 > x_max)
   x1 = x_max;
end
if (x1 < x_min)
   x1 = x_min;
end

x2_dot = x3;
x3_dot = -rho * g * Ad / M * x1 - 1/2 * rho * Ao / M * Cd * x3 * abs(x3);  % equation 9
x_out = [x1_dot;x2_dot;x3_dot];
```

Comand Line Input to run the program:

```
>> [t,x]=ode45('candidate_3',[0 20],[0 0 0]);

>> figure(1)

>> subplot(311)
>> plot(t, x(:,1)*AreaDiaphram);
>> grid
>> ylabel ('Volume Displaced [m^3]');

>> subplot(312)
>> plot(t,-x(:,2));
>> grid
>> ylabel ('Vehicle Depth [m]');

>> subplot(313)
>> plot(t,-x(:,3));
>> grid
>> ylabel ('Vehicle Velocity [m/s]');

>> xlabel ('Time [s]');
```

```
%---------------------------------------------------------
% Buoyancy Engine Simulator
% Candidate 4 - Linear Actuator with internal hydraulic cylinder
% Nick Janes
% Feb 25 2004
% NRC-IOT
%---(Description)--------------------------------------
% This is the code used for the simulation in
% Section 3.4.4. The command line input is
% located at the end of the script.
%---------------------------------------------------------
function x_out = candidate_4(t,x)

% constants ---------------------------------------------
inch_2_m = 0.0254;          % conversion factor from inch to meters [m/inch]
g = 9.81;                   % acceleration due to gravity [m/s^2]
rho = 1000;                 % density of water [kg/m^3]
D = 4 * inch_2_m;           % diameter of the cylinder [m]
L = 40 * inch_2_m;          % length of the cylinder [m]
Ao = (D^2)*pi/4;            % cross sectional area of cylinder [m^2]
V = Ao * L;                 % volume of the cylinder [m^3]
M = V * rho;                % mass of the cylinder (taking added mass to be zero) [kg]
Cd = 0.82;                  % coefficient of drag for cylinder in axial flow
Dp = 2 * inch_2_m;          % diameter of internal pistion [m]
Ap = (pi * Dp^2)/4;         % effective pressure area on internal piston [m^2]

% actuator parameters -----------------------------------
actuator_halfstroke = 4.90 * inch_2_m;      % halfstroke length [m]
actuator_speed = 3 * inch_2_m;              % actuator speed [m/s]

% actuator limits ---------------------------------------
x1d_max = actuator_speed;           % max actuator speed [m/s]
x1d_min = -actuator_speed;          % min actuator speed [m/s]
x_min = -actuator_halfstroke;       % negative halfstroke [m]
x_max = actuator_halfstroke;        % positive halfstroke [m]

x1 = x(1);          % actuator stroke [m]
x2 = x(2);          % vehicle position [m]
x3 = x(3);          % vehicle velocity [m/s]

% we have to limit the extension of the actuator
if (x1 >= 0 & x1 <= x_max)
   x1_dot = x1d_max;
elseif (x1 < 0 & x1 >= x_min)
   x1_dot = x1d_min;
else
```

```
   x1_dot = 0;
end

% make sure not to displace more then 250 cm^3 either way
if (x1 > x_max)
   x1 = x_max;
end
if (x1 < x_min)
   x1 = x_min;
end

x2_dot = x3;
x3_dot = -rho * g * Ap / M * x1 - 1/2 * rho * Ao / M * Cd * x3 * abs(x3); % equation 9
x_out = [x1_dot;x2_dot;x3_dot];
```

Comand Line Input to run the program:

```
>> [t,x]=ode45('candidate_4',[0 20],[0 0 0]);

>> figure(1)

>> subplot(311)
>> plot(t, x(:,1)*AreaPistion);
>> grid
>> ylabel ('Volume Displaced [m^3]');

>> subplot(312)
>> plot(t,-x(:,2));
>> grid
>> ylabel ('Vehicle Depth [m]');

>> subplot(313)
>> plot(t,-x(:,3));
>> grid
>> ylabel ('Vehicle Velocity [m/s]');

>> xlabel ('Time [s]');
```

```
%--------------------------------------------------------
% Buoyancy Engine Simulator
% Candidate 5 - Pumping Surrounding Fluid into Internal Bladder
% Nick Janes
% Feb 25 2004
% NRC-IOT
%---(Description)--------------------------------------
% This is the code used for the simulation in
% Section 3.4.5. The command line input is
% located at the end of the script.
%--------------------------------------------------------
function x_out = candidate_5(t,x)

% constants ----------------------------------------------
inch_2_m = 0.0254;          % conversion factor from inch to meters [m/inch]
g = 9.81;                   % acceleration due to gravity [m/s^2]
rho = 1000;                 % density of water [kg/m^3]
D = 4 * inch_2_m;           % diameter of the cylinder [m]
L = 40 * inch_2_m;          % length of the cylinder [m]
Ao = (D^2)*pi/4;            % cross sectional area of cylinder [m^2]
V = Ao * L;                 % volume of the cylinder [m^3]
M = V * rho;                % mass of the cylinder (taking added mass to be zero) [kg]
Cd = 0.82;                  % coefficient of drag for cylinder in axial flow

% pump constants -----------------------------------------
pump_speed = 6.825 * 10^(-5);          % flow speed [m^3/s]
pump_max_speed = pump_speed;           % max pump speed
pump_min_speed = -pump_speed;          % min pump speed

% bladder constants --------------------------------------
bladder_max = 0.000250;          % bladder upper limit [m^3]
bladder_min = -0.000250;         % bladder lower limit [m^3]

x1 = x(1);          % reset bladder contents [m^3]
x2 = x(2);          % reset vehicle depth [m]
x3 = x(3);          % reset vehicle velocity [m/s]

% limit overfilling of the bladders ----------------------
if (x1 >= 0 & x1 <= bladder_max)
   x1_dot = pump_max_speed;
elseif (x1 < 0 & x1 >= bladder_min)
   x1_dot = pump_min_speed;
else
   x1_dot = 0;
end
```

```
% make sure not to overfill past 250 cm^3
if (x1 > bladder_max)
   x1 = bladder_max;
end
if (x1 < bladder_min)
   x1 = bladder_min;
end

x2_dot = x3;
x3_dot = -rho * g * x1 / M - 1/2 * rho * Ao / M * Cd * x3 * abs(x3);        % equation 9
x_out = [x1_dot;x2_dot;x3_dot];
```

Comand Line Input to run the program:

```
>> [t,x]=ode45('candidate_5',[0 20],[0 0 0]);

>> figure(1)

>> subplot(311)
>> plot(t, x(:,1));
>> grid
>> ylabel ('Volume Displaced [m^3]');

>> subplot(312)
>> plot(t,-x(:,2));
>> grid
>> ylabel ('Vehicle Depth [m]');

>> subplot(313)
>> plot(t,-x(:,3));
>> grid
>> ylabel ('Vehicle Velocity [m/s]');

>> xlabel ('Time [s]');
```

```
%------------------------------------------------------
% Normalizing Velocities – Figure 15
% Nick Janes
% Feb 26 2004
% NRC-IOT
%----(Description)-----------------------------------
% The following command line input will
% plot Figure 15 in Section 3.5
%------------------------------------------------------

>> clear

>> vterm = 0.86;        % Terminal Velocity [m/s]

>> figure(1)

>> [t,x1]=ode45('candidate_1',[0 15],[0 0 0]);
>> plot (t,-x1(:,3)/vterm, 'bo-');
>> hold on

>> hold on

>> [t,x2]=ode45('candidate_2',[0 15],[0 0 0]);
>> plot (t,-x2(:,3)/ vterm, 'r--');
>> hold on

>> [t,x3]=ode45('candidate_3',[0 15],[0 0 0]);
>> plot (t,-x3(:,3)/ vterm, 'kv-');
>> hold on

>> [t,x4]=ode45('candidate_4',[0 15],[0 0 0]);
>> plot (t,-x4(:,3)/ vterm, 'ro-.');
>> hold on

>> [t,x5]=ode45('candidate_5',[0 15],[0 0 0]);
>> plot (t,-x5(:,3)/ vterm, ' bo-.');
>> hold on

>> AXIS([0 10 0 1.1])
>> grid
>> ylabel ('Normalized Vehicle Velocity [m/s]');
>> xlabel ('Time [s]');
>> legend('Candidate 1 & 5','Candidate 2', 'Candidate 3','Candidate 4')
```

```
%--------------------------------------------------------
% Normalizing Velocities – Figure 16
% Nick Janes
% Feb 26 2004
% NRC-IOT
%----(Description)-----------------------------------
% The following command line input will
% plot Figure 16 in Section 3.5
%--------------------------------------------------------

>> clear

>> vterm = 0.86;        % Terminal Velocity [m/s]

>> figure(1)

>> [t,x1]=ode45('candidate_1',[0 15],[0 0 0]);
>> plot (-x1(:,2),-x1(:,3)/0.86,'bo-');
>> hold on

>> [t,x2]=ode45('candidate_2',[0 15],[0 0 0]);
>> plot (-x2(:,2),-x2(:,3)/0.86,'r+-');
>> hold on

>> [t,x3]=ode45('candidate_3',[0 15],[0 0 0]);
>> plot (-x3(:,2),-x3(:,3)/0.86,'kv-');
>> hold on

>> [t,x4]=ode45('candidate_4',[0 15],[0 0 0]);
>> plot (-x4(:,2),-x4(:,3)/vterm,'ro-.');
>> hold on

>> [t,x5]=ode45('candidate_5',[0 15],[0 0 0]);
>> plot (-x5(:,2),-x5(:,3)/vterm,'bo-');
>> hold on

>> AXIS([0 3 0 1.05])
>> grid
>> ylabel ('Normalized Vehicle Velocity [m/s]');
>> xlabel ('Vehicle Position [m]');
>> legend('Candidate 1 & 5','Candidate 2', 'Candidate 3','Candidate 4')
```

```
%--------------------------------------------------
% gb_plot
% Nick Janes
% March 30 2004
% NRC-IOT
%---(Description)------------------------------
% Plots locations of the Center of Gravity
% and Center of Buoyancy relative to
% Linear Actuator position. See Figure 18 in
% Section 4.0.
%--------------------------------------------------
function gb_plot

%------------------------------------------------------------------------
% Find the Center of Gravity

%  extension of actuator from lower limit to upper limit
extension = [0.0243:0.0001:0.1459];

m_1 = 1.134;            % mass of the actuator static base [kg]
L_1 = 0.082042;         % length of the actuator static base [m]
m_2 = 0.340;            % mass of the actuator static tube [kg]
L_2 = 0.26289;          % length of the actuator static tube [m]

% center of gravity of the static part of the actuator
% measured from the base of the actuator
G_1 = (0.5*m_1*L_1 + m_2*(L_1 + 0.5*L_2))/(m_1 + m_2);

m_3 = 0.227;            % mass of the moving rod in the actuator [kg]
L_3 = 0.21209;          % length of the moving rod in the actuator [m]
m_4 = 1.9742;           % mass of the piston mounted on the actuator [kg]
L_4 = 0.06604;          % length of the piston [m]

% center of gravity of the dynamic part of the actuator
% measured from the base of the dynamic part
G_2 = (0.5*m_3*L_3 + (L_3 + 0.5*L_4)*m_4)/(m_3 + m_4);

% center of gravity of the actuator measured from its base
% as a function of actuator extension
for i = 1:1217
   G_3(1,i) = (G_1*(m_1 + m_2) + (L_1 + L_2 + extension(1,i) - (L_3 - G_2))*(m_3 +
m_4))/(m_1 + m_2 + m_3 + m_4);
end

% plot the Actuator Position vs. Center of Buoyancy Curve
figure(2);
```

```
plot (extension,G_3,'r');
xlabel('Actuator Longitudinal Position [m]');
ylabel('Longitudinal Position of Center of Gravity of Actuator[m]');
grid;

m_5 = 0.0903;          % mass of the diaphragm tightening ring [kg]
L_5 = 0.0127;          % distance to the center of mass m5 [m]
m_6 = 0.227;           % mass of the diaphragm support [kg]
L_6 = 0.05842;         % distance to the center of mass m6 [m]
m_7 = 0.71035;         % mass of the electronics module [kg]
L_7 = 0.75720;         % distance to the center of mass of the electronics module [m]
m_8 = 0.5863;          % mass of the endcap [kg]
L_8 = 0.9692;          % distance to the center of mass of the endcap [m]
m_9 = 2.3009;          % mass of the PVC pipe section [kg]
L_9 = 0.4873;          % distance to the center of mass of the pipe section [m]
m_10 = 2.32;           % mass of the batteries [kg]
L_10 = 0.6683;         % distance to the center of mass of the batteries [m]
m_11 = 0.1369;         % mass of the actuator mounting plate [kg]
L_11 = 0.5254;         % distance to the center of mass of the mounting plate [m]
m_12 = 0.068;          % mass of the mounting plate supports [kg]
L_12 = 0.5540;          % distance to center of mass of the mounting plate supports [m]

M_total =       m_1 + m_2 + m_3 +
                m_4 + m_5 + m_6 +
                m_7 + m_8 + m_9 +
                m_10 + m_11 + m_12;

Static_Part =   m_5*L_5 + m_6*L_6 + m_7*L_7 +
                m_8*L_8 + m_9*L_9 + m_10*L_10 +
                m_11*L_11 + m_12*L_12;

Dynamic_M = m_1 + m_2 + m_3 + m_4;


for i = 1:1217
   G(1,i) = ((0.5222494 - G_3(1,i))*Dynamic_M + Static_Part)/(M_total);
end

% plot the Actuator Position vs. Center of Buoyancy Curve
figure(1);
plot (extension,G,'r');
xlabel('Actuator Longitudinal Position [m]');
ylabel('Longitudinal Position of Center of Gravity [m]');
grid;
```

DESIGN OF A BUOYANCY ENGINE FOR AN UNDERWATER GLIDER

```
%------------------------------------------------------------------------
% Find the Center of Buoyancy

%  x (row, column)
x = [-0.0608:0.0001:0.0608];

% calculate L1 and V1
L_1 = 0.5125;
V_1 = 0.01;

% calculator L2 and V2
L_2 = 0.025 / 2;
V_2 = 1.425*10^(-4);

% loop through x vector calculating
% values for the other vectors
for i = 1:1217

   % calculate L3 and V3 vectors
   L_3(1,i) = 0.025 - (x(1,i) / 2);
   V_3(1,i) = ( 3.66*10^(-3) ) * x(1,i);

   % calculate L4 and V4 vectors
   L_4(1,i) = 0.025 + 0.01556 - (0.25 * x(1,i));
   V_4(1,i) = (x(1,i) * 4.483*10^(-5)) - 2.79*10^(-5);

   % calculate the total volume vector
   V_total(1,i) = V_1 + V_2 + V_3(1,i) + V_4(1,i);

   % calculate the center of buoyancy vector
   L_c(1,i) =  (L_1 * V_1 + L_2 * V_2 + L_3(1,i) * V_3(1,i) + L_4(1,i) * V_4(1,i)) /
V_total(1,i);

end

% plot the Actuator Position vs. Center of Buoyancy Curve
figure(3);
plot (x,L_c,'r');
xlabel('Actuator Longitudinal Position Relative B [m]');
ylabel('Longitudinal Position of Center of Buoyancy Relative A [m]');
grid;
```

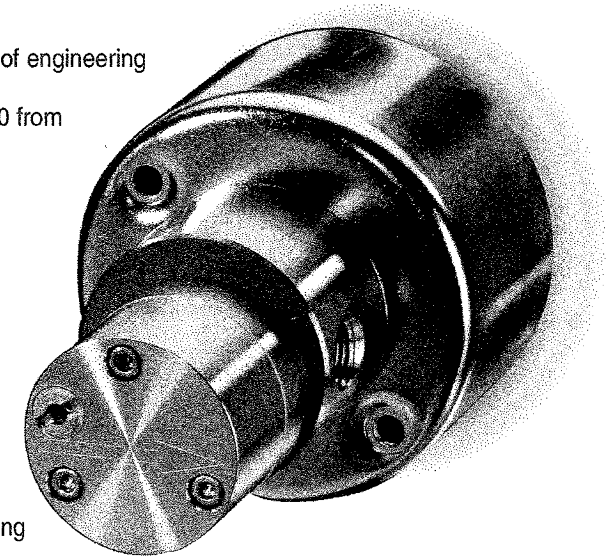
```
%------------------------------------------------------------------------
% Plot Center of Gravity and Center of Buoyancy vs. Actuator position

figure(4);
plot (x,L_c,'r',x,G,'b');
xlabel('Actuator Longitudinal Position Relative B [m]');
ylabel('Positions of Center of Buoyancy and Center of Gravity [m]');
grid;
```

APPENDIX B: DATASHEETS

SERIES 120

Magnetic Drive Gear Pump



Series 120 Backed by a tradition of engineering and technological expertise, the Series 120 from Micropump delivers exceptional pumping performance for any high precision application. These compact, magnetically driven gear pumps feature a cavity style design with Teflon® seals to ensure leak-free performance. With benefits like chemical resistance, abrasive fluids pumping and smooth, pulseless delivery, Series 120 pumps are available with a wide range of options, as well as in standard and OEM configurations.

Small Size
The miniature package size of the Series 120 delivers performance that is usually found only in much larger pumps.

Leak-Free
The magnetic drive and Teflon seals keep the fluid securely inside the pump and potential contaminants out.

Smooth Pulseless Delivery
Positive displacement, helical gears provide smooth, accurate fluid delivery that is proportional to motor speed.

Chemically Resistant
An extensive range of wetted material options ensure compatibility with a wide range of chemicals.

Easy to Service
Series 120 pumps are easy to service using a Micropump service kit and simple hand tools.

Wide Range of Options
Product design offers the flexibility to configure the product to your specific needs. Some options include:

- Three standard gear sizes
- Multiple gear and body materials
- Optional internal bypass
- Optional high torque magnets
- Side or deck port configurations
- NEMA, IEC, and Micropump drive mounts

Advanced Pump Technology
Micropump uses the latest engineering tools and manufacturing equipment to produce the most advanced pumping solutions available. Products are developed using state-of-the-art CAD, Finite Element Analysis (FEA), and rapid prototyping tools. Precision CMM and CNC manufacturing equipment ensure the highest level of product quality.

Proven Reliability
Nearly 40 years of experience solving the most difficult pumping problems go into the design and manufacture of every Micropump pump, ensuring the most reliable pumping solution available.

OEM Configurations
Series 120 pumps can be customized to meet your individual requirements.

Enhanced Efficiency
As part of the IDEX family of companies, Micropump utilizes Kaizen, Lean Manufacturing and Six Sigma process improvement strategies to continually meet the challenge of improving quality while increasing productivity—all of which help Micropump better meet the pumping needs of customers in a increasingly diverse range of markets.

MICROPUMP

IDEX CORPORATION

SERIES 120

Magnetic Drive Gear Pump

GJ

Performance Summary

Flow Rate
at 3450 rpm

3.2 L/min			0.85 gpm
Displacement (ml/rev)			
Gear Set	N21	N23	N25
ml/rev	0.316	0.64	0.91
Maximum Differential Pressure			
5.6 Bar			80 psi
Maximum System Pressure			
21 Bar			300 psi
Temperature Range			
-46 to 121° C			-50 to 250° F
Viscosity Range			Maximum Speed
0.2 to 1500 cps			10,000 rpm

Pump Construction

- Magnetic drive gear pump
- Cavity style
- Two helical, shafted gears
- Sleeve bushings
- Teflon® gaskets and seals

Wetted Materials

- Base material
  - 316 stainless steel
  - Alloy 20
  - Titanium
  - Hastelloy C-276®
  - Hastelloy B-2®
  - Ceramics (Zirconia)
- Gears
  - PEEK
  - Ryton® (PPS)
  - Teflon
  - Nickel-Carbide
- Static seals
  - Teflon

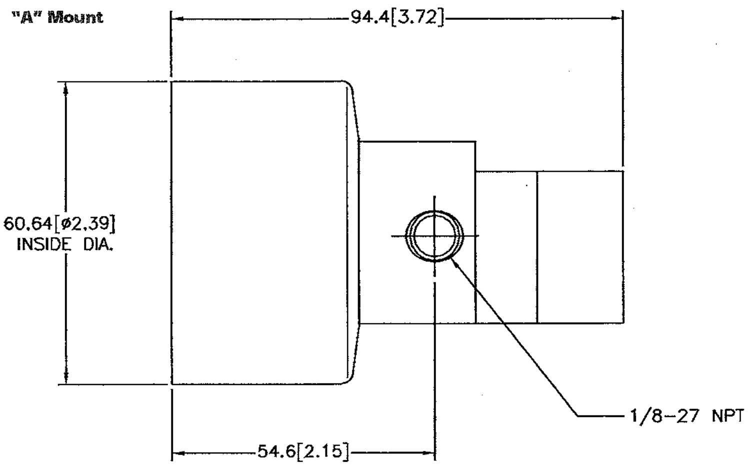
Magnets

- Driven and driving
  - Ferrite
  - Rare earth

Product Enhancements

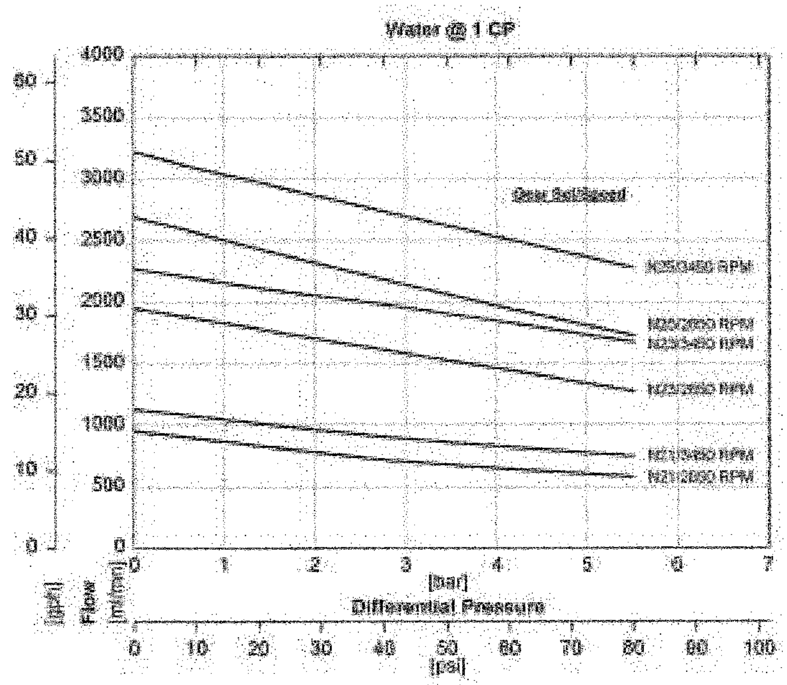
- Internal bypass
- Deck ports
- Tri-clamp fittings
- Carbon bushings

Dimensions

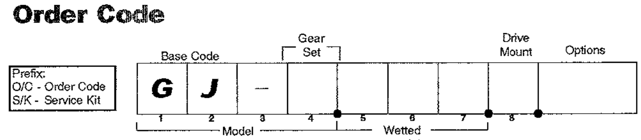


Units: mm[in]. Nominal dimensions shown.

Pump Performance



Order Code



ACTUAL PERFORMANCE MAY VARY

Specifications are subject to change without notice.

TT2746MPV-05/2001

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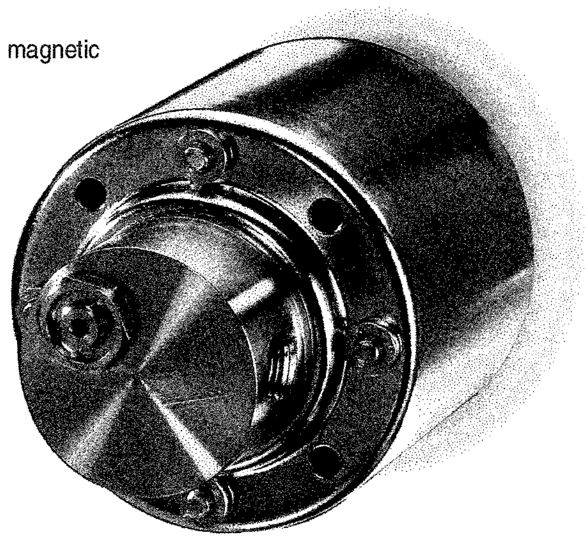
MICROPUMP

IDEX CORPORATION

SERIES 200

Magnetic Drive Gear Pump

Series 200 Micropump's Series 200 magnetic drive gear pump features a patented suction shoe design that delivers the high performance needed to keep your operations flowing smoothly. The unique design self compensates for wear, ensuring near zero slip and a longer pump life. Available in standard or custom configurations, Series 200 pumps offer the flexibility to meet a wide range of applications with a compact size, options like two standard gear sizes, an internal bypass, and variety of materials.



Small Size
The miniature package size of the Series 200 delivers performance that is usually found only in much larger pumps.

Leak-Free
The magnetic drive and a single O-ring seal keep the fluid securely inside the pump and potential contaminants out.

Smooth Pulseless Delivery
Positive displacement, helical gears provide smooth, accurate fluid delivery that is proportional to motor speed.

Chemically Resistant
An extensive range of wetted material options ensure compatibility with a wide range of chemicals.

Easy to Service
Series 200 pumps are easy to service using a Micropump service kit and simple hand tools.

OEM Configurations
Series 200 pumps can be customized to meet your individual requirements.

Wide Range of Options
Product design offers the flexibility to configure the product to your specific needs. Some options include:

- Two standard gear sizes
- Two and three gear versions
- Multiple gear, body, and O-ring materials
- Optional internal bypass
- Two standard port sizes
- Optional high torque magnets
- NEMA, IEC, and Micropump drive mounts

Advanced Pump Technology
Micropump uses the latest engineering tools and manufacturing equipment to produce the most advanced pumping solutions available. Products are developed using state-of-the-art CAD, Finite Element Analysis (FEA), and rapid prototyping tools. Precision CMM and CNC manufacturing equipment ensure the highest level of product quality.

Proven Reliability
Over 40 years of experience solving the most difficult pumping problems go into the design and manufacture of every Micropump pump, ensuring the most reliable pumping solution available.

Enhanced Efficiency
As part of the IDEX family of companies, Micropump utilizes Kaizen, Lean Manufacturing and Six Sigma process improvement strategies to continually meet the challenge of improving quality while increasing productivity—all of which help Micropump better meet the pumping needs of customers in an increasingly diverse range of markets.

MICROPUMP

IDEX
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SERIES 200

Magnetic Drive Gear Pump

Performance Summary

Flow Rate
at 3450 rpm

4.0 L/min			1.1 gpm
Displacement (ml/rev)			
Gear Set	P23	P25	P35
ml/rev	0.26	0.58	1.17
Maximum Differential Pressure			
8.7 Bar			125 psi
Maximum System Pressure			
21 Bar			300 psi
Temperature Range			
-46 to 177° C			-50 to 350° F
Viscosity Range			
0.2 to 1500 cps			
Maximum Speed			
10,000 rpm			

Pump Construction

- Magnetic drive gear pump
- Suction shoe style
- Two or three helical gears
- Stationary shafts
- O-ring seal

Wetted Materials

- Base material
  - 316 stainless steel
  - Alloy 20
  - Titanium
  - Hastelloy C-276®
  - Hastelloy B-2®
- Gear material
  - PEEK
  - PPS
- Static seals
  - Viton®
  - EP
  - Buna N
  - Kalrez®

Magnets

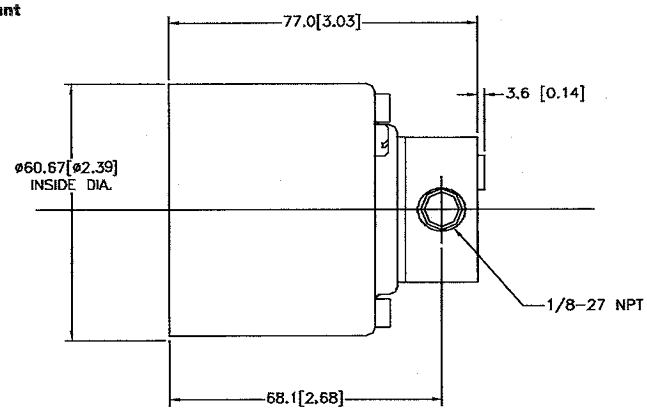
- Driven and driving
  - Ferrite
  - Rare earth

Product Enhancements

- Internal bypass
- 1/4-18 NPT ports
- Tri-clamp fittings

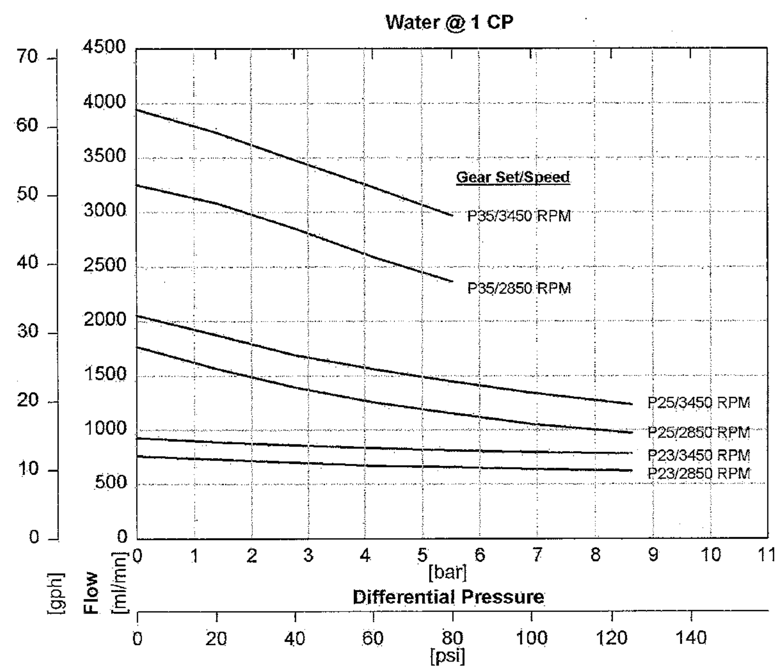
Dimensions

"A" Mount



Units: mm[in]. Nominal dimensions shown.

Pump Performance



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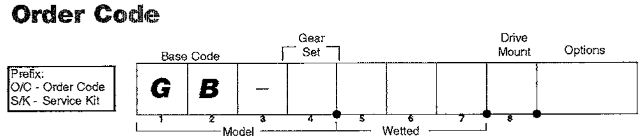

ACTUAL PERFORMANCE MAY VARY

ATEX certified Group II Category 2GD products are available.

Specifications are subject to change without notice.

MP1000-07/2003

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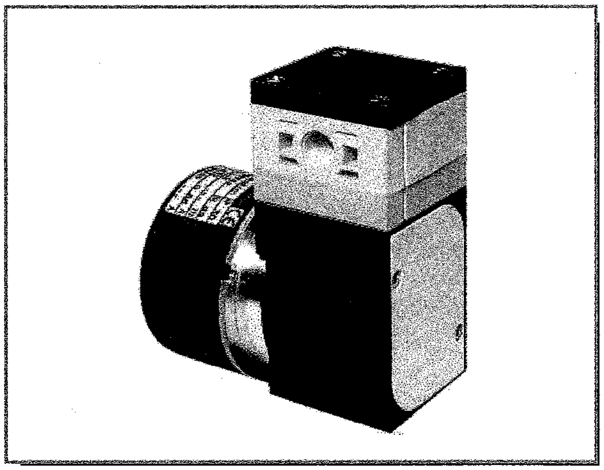
MICROPUMP

IDEX CORPORATION



Model NF100 shown with BLDC Motor

SECTION 200.60

Type: NF100, NF1.100

Mini-Diaphragm Liquid Pump

Corrosion-Resistant OEM Installation Models with AC, DC or BLDC Motors

CE

Flow Rate per Head (NF1.100):	1.3 liter/min. (20.6 GPH)
Suction Head/Vacuum:	13 ft. H_2O (12 in. Hg)
Max. Continuous Pressure Head (NF1.100):	197 ft. H_2O (85 psig)

Description

KNF's NF series of liquid diaphragm pumps are self-priming, can pump air, gases and liquids and simultaneously create a vacuum. Their unique, one-piece molded diaphragm is usually found only in larger, more expensive pumps. Metal wetted-parts are eliminated and end-vacuum performance is enhanced. A wide choice of materials, including PVDF, are available for corrosion-resistance where needed.

They are ideal for use in portable, AC or battery-operated equipment where high performance, low power consumption, minimal weight and size are important. The NF1.100, similar in construction to the NF100, provides a higher head pressure.

Pump Features

- **Flexible OEM Design** - KNF's engineers combine high performance with a small physical package to produce an efficient, compact unit. Optimally placed mounting holes in the compressor housing permit installation of this pump in any position. KNF's Project Pump program allows for a variety of inexpensive modifications to match your requirements.
- **High Chemical Resistance, Contamination-Free** - The use of chemically-resistant materials such as PTFE, PVDF, FFPM or other material combinations in the wetted parts, facilitates the pumping of almost all neutral or corrisive liquids. The TT model, with its PVDF/FFPM/PTFE wetted parts, is excellent for pumping severe corrosives or for use as a sampling pump in sensitive analyzers where low elastomer outgassing is important.
- **Self Priming** - These pumps run dry continuously without damage. They can pump air/liquid slurries, create wet-vacuum conditions, and transport air or gases.
- **Low Noise Level** - An enclosed compressor housing minimizes noise transmission and keeps dirt away from critical components.
- **Variety of Accessories** - Accessories include diaphragm pressure control valves, check valves and hose connectors.

Selected Applications

- Blood Analyzer Waste Removal
- Blood/Food Protein Analyzers
- Air/Gas Analyzers
- Wastewater Analyzers
- Ink Jet Systems
- Dispensing Systems
- Endoscopy Systems
- Immuno-Analytical Systems
- Photography Developers
- Titration Calorimeters
- Industrial Washing Machines
- Pipette, Probe & Needle Washers
- Transfer of Corrosive Liquids
- Filtration/Chromatography

NF100, NF1.100

KNF Performance Specifications

Model Number	NF100KPE	NF100KPDC	NF1.100KPE	NF1.100KPDC
Head Configuration	Single Head	Single Head	Single Head	Single Head
Maximum Flow	1.2 liters/min. (19 GPH)		1.3 liter/min. (20.6 GPH)	
Maximum Suction Head	13.1 ft. H_2O / 12 in. Hg / 4 mWg		13.1 ft. H_2O / 12 in. Hg / 4 mWg	
Maximum Continuous Pressure Head	33 ft. H_2O / 15 psig / 10 mWg		197 ft. H_2O / 85 psig / 60 mWg	

Electrical

Motor Type	Shaded Pole AC	Brush type DC	Shaded Pole AC	Brush type DC
Motor Protection	IP00	IP50	IP00	IP50
Motor Voltage (frequency)	115 VAC (60 Hz)	6 / 12 / 24	115 VAC (60 Hz)	6 / 12 / 24
Full Load Motor Current	0.8 Amps	1.7 / 0.9 / 0.3	0.85 Amps	2.7 / 1.3 / 0.5

Environmental

Maximum Ambient Temperature	+40°C (105°F) All models			
Maximum Medium Temperature	-10°C to +80°C (+14°F to +176°F) (Models available to 135°C.)			
Net Weight	1100 gr (2.5 lbs.)	600 gr (1.3 lbs.)	1600 gr (3.6 lbs.)	720 gr (1.6 lbs.)
EMV Guideline	EN55014	EN55014	EN55014	EN55014

Materials of Construction

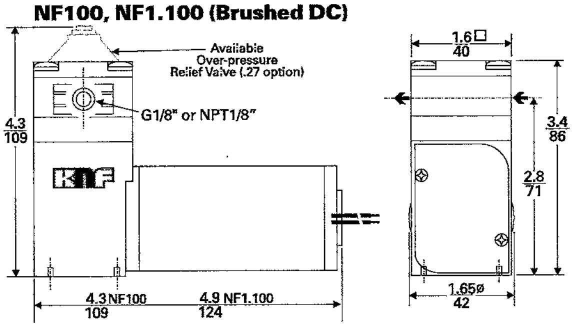
Material Code	Head	Diaphragm	Valves	O-Rings
NF100KP / NF1.100KP	Polypropylene	PTFE	EPDM	EPDM
NF100KT / NF1.100KT	Polypropylene	PTFE	FFPM	PTFE
NF100TT / NF1.100TT	PVDF	PTFE	FFPM	PTFE

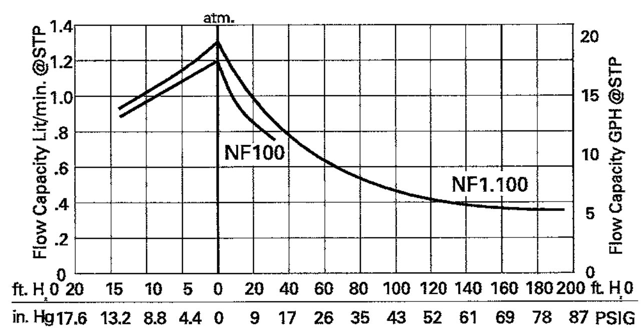
Notes: Standard continuous performance ratings listed above are per head, with water at 70°F (20°C) thru 8 mm tubing, and nominal electric supply. Dimensions and performance characteristics given are for reference only and are subject to change without notice. Higher performance models, various motor options including brushless DC and different materials of construction are available. Teflon® and Kalrez® are registered trademarks of DuPont.

Accessories: Integrated, Adjustable Over-Pressure Relief Valve, Brushless DC motor, Air-Driven Motor, Pulsation Dampener, Twin-Head Models, Hose Connectors, Elbow Fittings, Shock Mounts

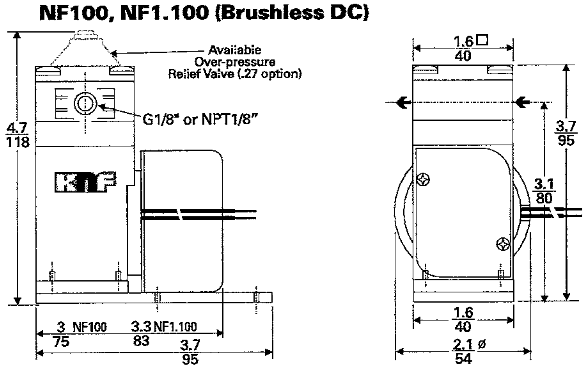
Performance Characteristics/Outline Dimensions:

NF100, NF1.100 (Brushed DC)

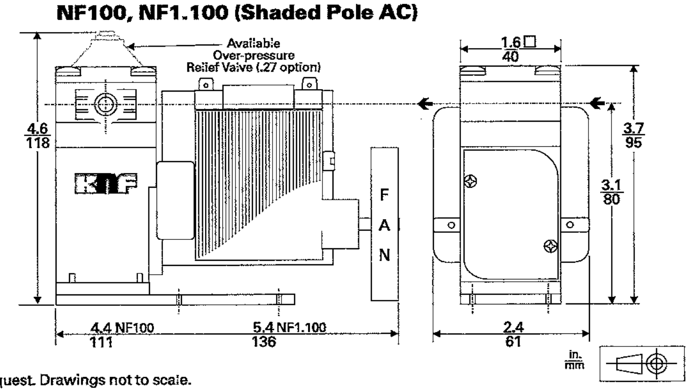




NF100, NF1.100 (Brushless DC)

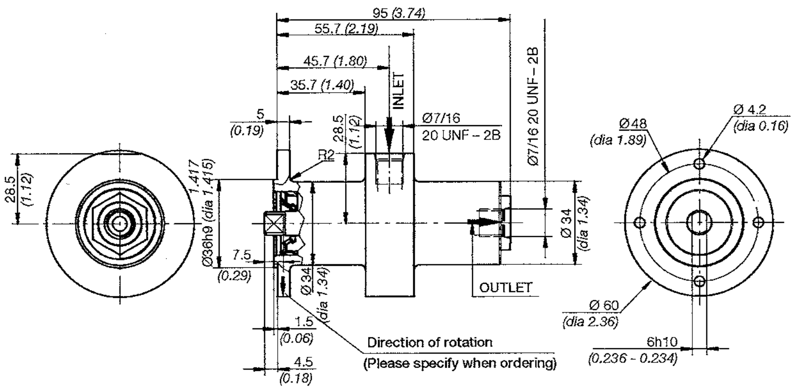


NF100, NF1.100 (Shaded Pole AC)



Actual motor appearance and dimensions may vary. Certified drawings are available on request. Drawings not to scale.

KNF NEUBERGER, INC.
Two Black Forest Road
Trenton, New Jersey 08691-1810
Phone: 609-890-8600 · Fax: 609-890-8323
Web: http://www.knf.com • pumps@knf.com



Dimensions are written in millimeters and *(inches)* .

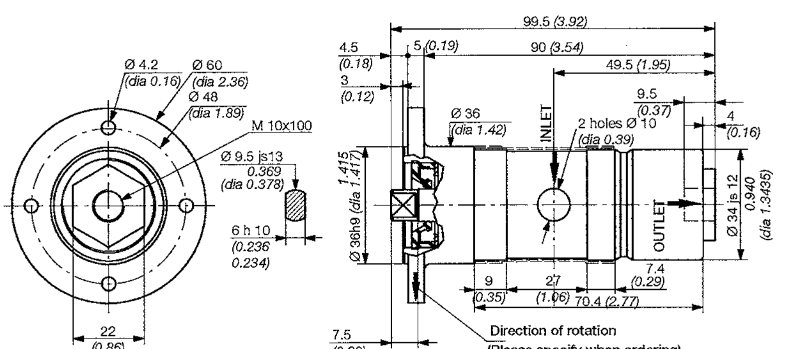
Pump		Displacement (cu.in/rev) *(cm³/rev)*	Max. pressure continuous (PSI) / *(bar)*	Max. pressure peak (PSI) / *(bar)*	Max. speed continuous (rpm)	Max. speed peak (rpm)
PB36.5	0511320	0.0070	4350	5080	5000	6000
		0.115	*300*	*350*		
PB36.5	0511280	0.0170	4350	5080	5000	6000
		0.280	*300*	*350*		
PB36.5	0511330	0.0220	4350	5080	5000	6000
		0.360	*300*	*350*		

PB 36.5
fixed displacement pump with external inlet adaptor

Displacement:
from 0.0070 to 0.0220cu.in/rev
(0.115 to 0.360cm³/rev)
5080PSI (350bar)
5000rpm
3 pistons

Maximum operating temperature: 392°F (200°C)
Weight: 1.98lbs (0.9kg)

The following may be modified on request:
- input and outlet ports,
- mounting flange or thread,
- drive shaft.



Dimensions are written in millimeters and *(inches)*.

Pump		Displacement (cu.in/rev) *(cm³/rev)*	Max. pressure continuous (PSI) / *(bar)*	Max. pressure peak (PSI) / *(bar)*	Max. speed continuous (rpm)	Max. speed peak (rpm)
PB36.5	050720	0.0070	4350	5080	5000	6000
		0.115	*300*	*350*		
PB36.5	050790	0.0170	4350	5080	5000	6000
		0.280	*300*	*350*		
PB36.5	057310	0.0220	4350	5080	5000	6000
		0.360	*300*	*350*		

PB 36.5
fixed displacement pump for use submerged in oil

Displacement:
from 0.0070 to 0.0220cu.in/rev
(0.115 to 0.360cm³/rev)
5080PSI (350bar)
5000rpm
3 pistons

Maximum operating temperature: 392°F (200°C)
Weight: 1.32lbs (0.6kg)

The following may be modified on request:
- input and outlet ports,
- mounting flange or thread,
- drive shaft,
- addition of a filter/screen.

PB 36.5

Fixed displacement pumps

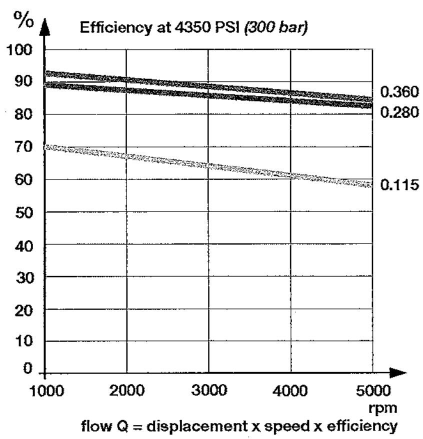
PB 36.5 standard

pump	direction of rotation	part number	displacement (cu.in/rev) *(mm³/rev)*	outlet	inlet	shaft	Max. Pressure (PSI) *(bar)*
PB 36.5	CW	050790	0.0017 *280*	M10 x 100	2x0.39 (2xØ10)	6/flat	5080 *350*
PB 36.5	CCW	050720	0.0070 *115*	M10 x 100	2x0.39 (2xØ10)	6/flat	5080 *350*
PB 36.5	CCW	057310	0.0220 *360*	M10 x 100	2x0.39 (2xØ10)	6/flat	5080 *350*
PB 36.5	CW	057315	0.0220 *360*	M10 x 100	2x0.39 (2xØ10)	6/flat	5080 *350*
PB 36.5	CW	050640	0.0170 *280*	7/16-20 UNF - 2B	2x0.39 (2xØ10)	6/flat	5080 *350*
PB 36.5	CCW	0510640	0.0220 *360*	M10 x 100	2x0.39 (2xØ10)	key shaft	5080 *350*
PB 36.5	CW	0511280	0.0170 *280*	7/16-20 UNF - 2B	7/16-20 UNF - 2B	6/flat	5080 *350*
PB 36.5	CW	0511585	0.0103 *170*	3/8- 24 UNF	2x0.39 (2xØ10)	6/flat	5080 *350*
PB 36.5	CCW	0512490	0.0220 *360*	7/16-20 UNF - 2B	2x0.39 (2xØ10)	6/flat	5080 *350*
PB 36.5	CW	057920	0.0092 *150*	3/8- 24 UNF	2x0.39 (2xØ10)	6/flat	5080 *350*

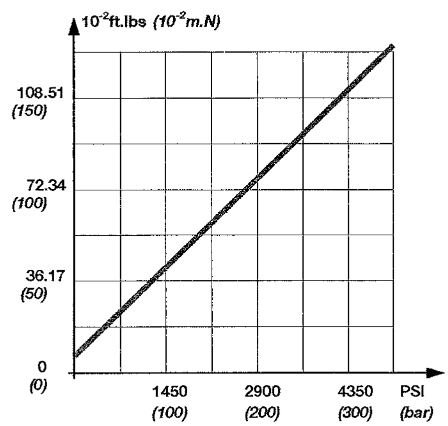
Operating conditions
Be careful to respect filtration/fluid cleanliness conditions in particular, see page 21.

Fluid
Fluid viscosity should, at extreme operating temperatures, be between 1cSt (1°E) and 300cSt (40°E).
For other requirements, please consult our Technical Department.

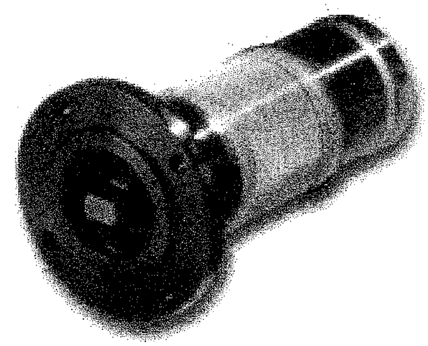
*Volumetric efficiency**



*Torque**



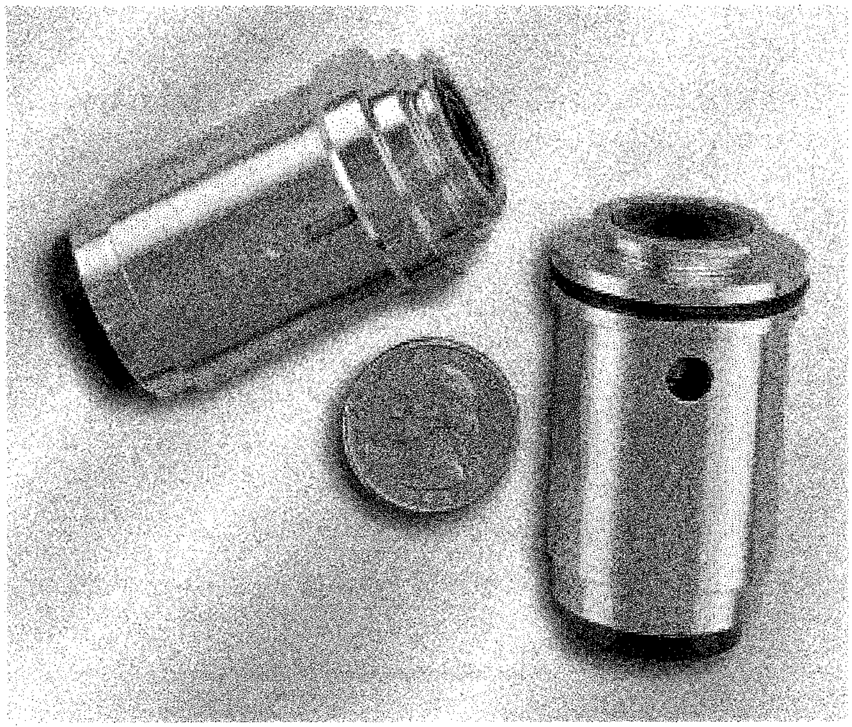
*Graphs are given as an indication only, for calculation formula see page 17.



HYDRO LEDUC

Compact Fluid Power Redefined by the new Oildyne Cartridge Piston Pump

The new Oildyne cartridge pump raises the standard for compact fluid power! This three-piston cartridge pump is an efficient, fixed-displacement pump that provides high performance at a very economical price. Pressure ratings of up to 276 bar (4000 psi), driven speeds of up to 5000 rpm, and the ability to provide a variety of seal types make this the solution to your unique applications. The uni-directional pump is capable of pumping fluids ranging in viscosity from solvents to thick fluids.

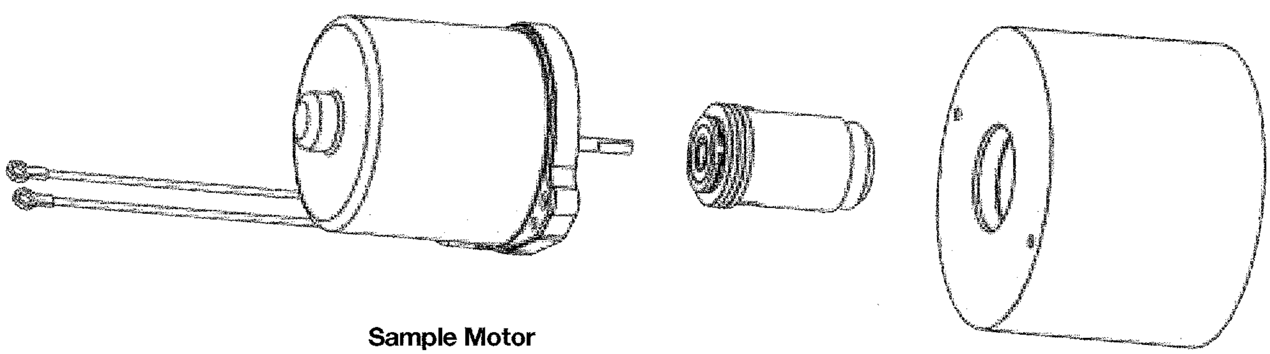


(ACTUAL SIZE PHOTO)

The three-piston cartridge pump maintains the performance and flexibility of the Oildyne five-piston, stand-alone pump while reducing the overall package dimensions.

This ultra-compact piston pump, approximately 33 mm (1.3") in diameter and 51 mm (2") long, is designed to fit into your specially machined manifold allowing for a custom package that fits your space needs.

A variety of displacements can be produced all within the existing physical size. (The internal cam angle determines the displacement.)



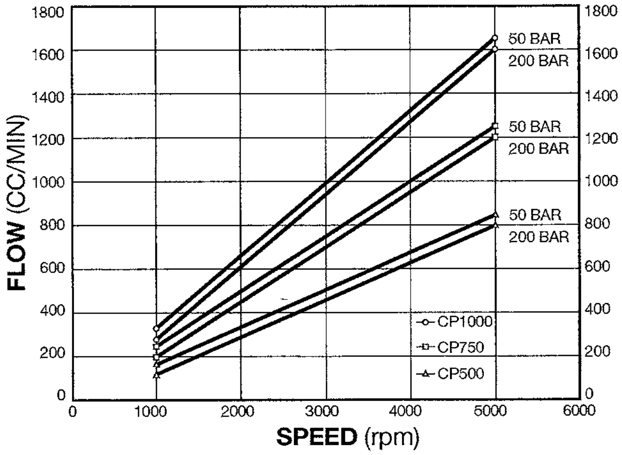
Sample Motor

Pump

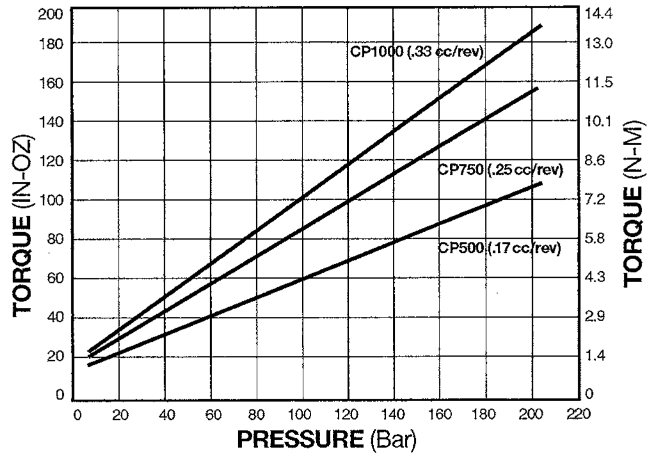
Your Manifold Package

Representative Performance Characteristics

Cartridge pump flow at 23°C on DOT 3 brake fluid



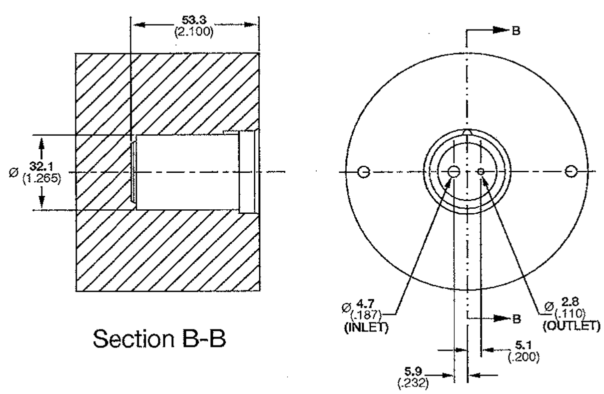
Cartridge pump input torque at 23°C on DOT 3 brake fluid



Cartridge Pump Dimensions

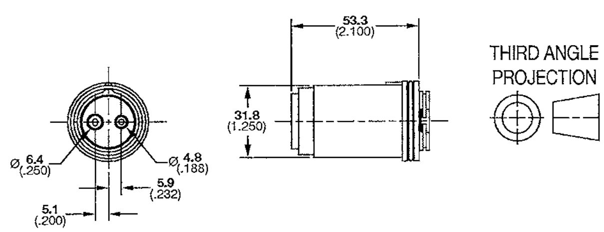
(with sample manifold requirements)

*All dimensions in **mm** (inches)*



Section B-B

Sample Manifolding Requirements



Cartridge Dimensions
(All Displacements)

Specifications

Displacements: .1 cc/rev. (.006 in³/rev.) to .33 cc/rev. (.020 in³/rev.)

Speeds: Up to 5000 rpm maximum

Pressures: 207 bar (3000 psi) maximum continuous
276 bar (4000 psi) maximum intermittent

Temperature Ranges: Up to 120°C (250°F)

Seals Available: Variety

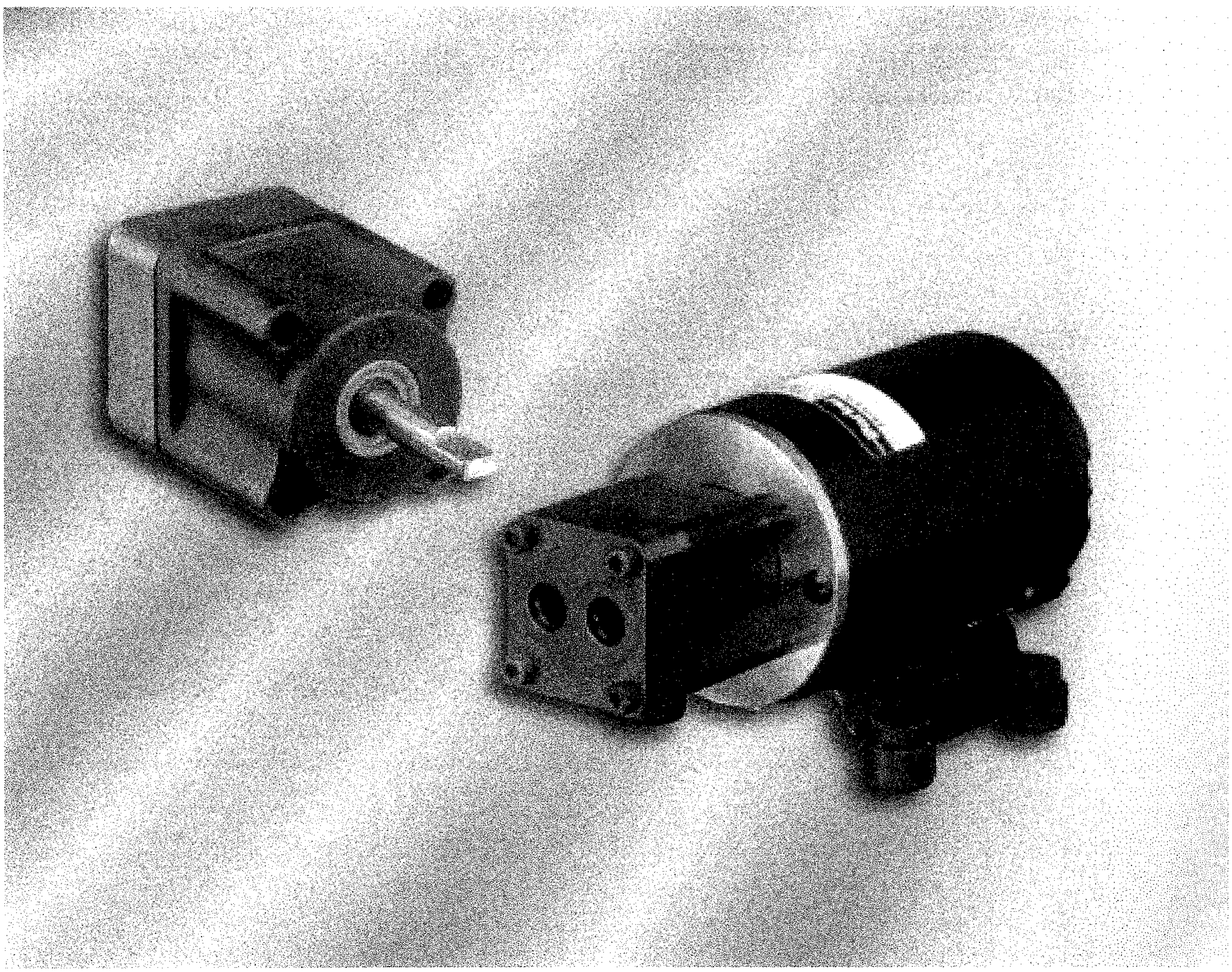
Fluids Compatibility: Variety

Specifications subject to change without notice.
Performance data is for reference only.

The new cartridge piston pump continues Oildyne's tradition of producing innovative products which can be customized to specific industries. Please call us to discuss how this cartridge pump can be used in your unique application.

Miniature Piston Pumps
5 Piston Design

Pressures to 276 bar (4000 psi)
Displacements from .156cc/rev
to .865cc/rev (.01 to .05 in³/rev)



Pumping Efficiencies to 90% Allow You to Effectively Use .156 to .865 cc Flow Per Rev. at Pressures to 276 bar (4000 psi)

Once in a great while there's a breakthrough design whose versatility opens broad new opportunities. Oildyne's mini pumps are a prime example.

Mini pumps pump or meter hydraulic oil, brake fluid, and Mil 5606 with equal ease. Need greater versatility?

These fixed displacement axial piston pumps are efficient and powerful too. Tests run on 78 SUS viscosity fluid at 100°F @ 3000 psi showed 90 percent volumetric efficiency. Capable of 276 bar (4000 psi) operation, mini pumps are available in nine model sizes from .156 to .865 cc per revolution displacement.

Compact size, versatility, efficiency, power and speed are quietly combined in a very cost competitive package in Oildyne's mini pumps. They're suitable for most applications requiring compact power including automotive, marine, medical and military uses.

Mini Pump Features

- .156 to .865 cc displacement per revolution.
- Designed for open circuit systems
- Fixed displacement - Output is controlled by motor speed
- Operating temperature range -40°C to +149°C (-40°F to +300°F).
- Naturally aspirated to 5000 rpm and above depending upon viscosity
- Porting on sides or rear
- Will operate efficiently on extremely thin (1 cS) fluid
- Multiple pumps, special configurations and bi-directional pumps are available on special order.

General Specifications

Model	156	206	259	311	346	417	519	692	865
Displacement									
In^3 per rev.	.0095	.0126	.0158	.0190	.0211	.0255	.0317	.0422	.0527
cc /rev	.156	.206	.259	.311	.346	.417	.519	.692	.865
GPM @ 3000 RPM	.123	.163	.205	.247	.274	.330	.411	.548	.685
cc/min @ 3000 RPM	467	618	778	934	1038	1252	1557	2076	2590
Max RPM @ rated pressure W/O supercharge	4400	4200	4000	3800	3800	3700	3700	3600	3500
Operating Pressure (psi)									
Continuous	3500	3500	3500	3500	3500	3500	3500	3250	3000
Intermittent	3750	3750	3750	3750	3750	3750	3750	3500	3500
Maximum	4000	4000	4000	4000	4000	4000	4000	3750	3500

Parker Hannifin Corporation
Oildyne Division
Minneapolis, MN 55428 USA

Standard Product Ordering Code

H ☐ ☐ – ☐ – ☐ – ☐ ☐ – S

COMPONENT
H High Pressure Pump

ROTATION
R CW
L CCW
B Bi-rotational

CONFIGURATION
S Single pump
T Tandem pump

SINGLE or 1st PUMP SIZE

CODE	DISP.
156	.156 cc/rev
206	.206 cc/rev
259	.259 cc/rev
311	.311 cc/rev
346	.346 cc/rev
417	.417 cc/rev
519	.519 cc/rev
692	.692 cc/rev
865	.865 cc/rev

2nd PUMP SIZE

CODE	DISP.
000	Single pump
156	.156 cc/rev
206	.206 cc/rev
259	.259 cc/rev
311	.311 cc/rev
346	.346 cc/rev
417	.417 cc/rev
519	.519 cc/rev
692	.692 cc/rev
865	.865 cc/rev

SEALS
B Buna N
V Fluorocarbon Rubber
E EPR

SHAFT & MOTOR
OO Std. .81" shaft, no motor
OR 12 VDC, 1/3 hp, Perm. Mag., close coupled
OM 24 VDC, 1/3 hp, Perm. Mag., close coupled
AA 115 VAC, 3450 rpm, 1/3 hp, cap. start
AD 230 VAC, 3450 rpm, 1/3 hp, cap. start
CC 56C Kit, includes long shaft
CO Long shaft, no motor (for use with 56C kit)
RO No shaft, replacement pump for use only with close coupled DC motor
AO Pump and shaft ***only*** for use with standard 115 and 230 VAC motors above
ZZ Contact Factory for specials

PUMP BODY
S Standard

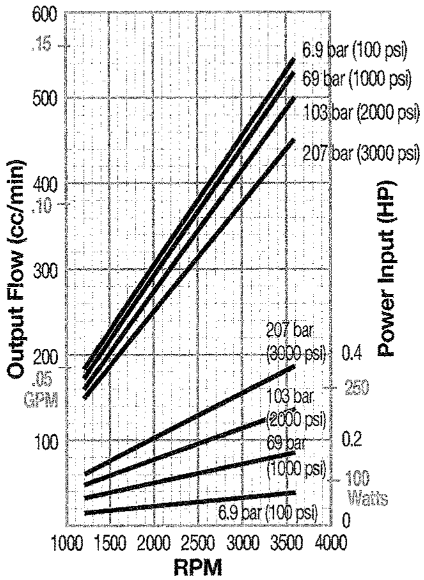
NOTES:
1. Tandem pumps must have larger displacement called out first
2. Tandem pumps are not available with the standard AC or DC motors - only plain shaft or 56C Kit
3. Drive shaft input torque must be under 3.5 n-m (525 in-oz) [equivalent to HRS865 operating at 207 bar (3000 psi); refer to catalog performance curves for torque data]
4. Bi-rotational pumps require the side port as case drain
5. For configurations not shown above please contact Oildyne

Performance Data

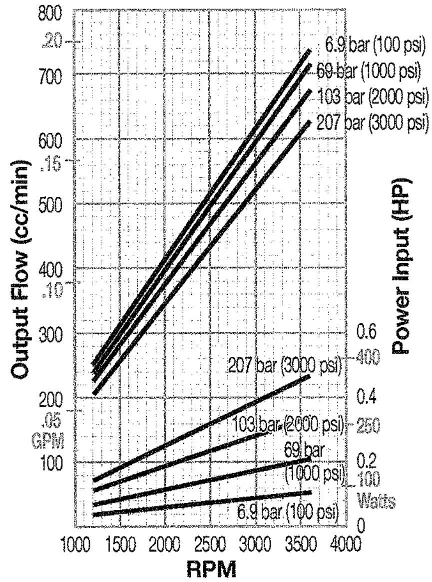
Performance data shown are the average results based upon a series of laboratory tests of production units and are not necessarily representative of any one unit. Tests were run with oil at 78 SUS at 38°C (100°F).

In accordance with our policy of continuing product development, we reserve the right to change specifications shown without notice.

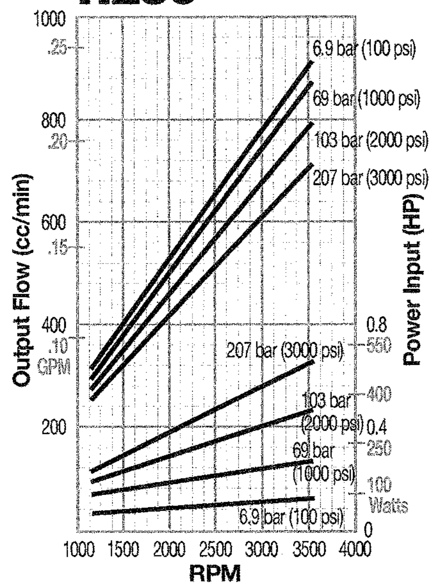
H156



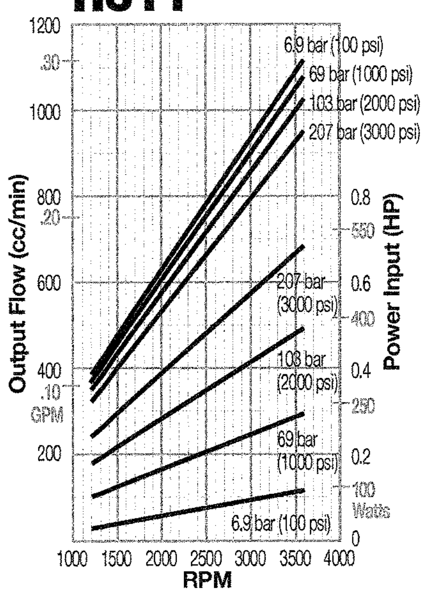
H206



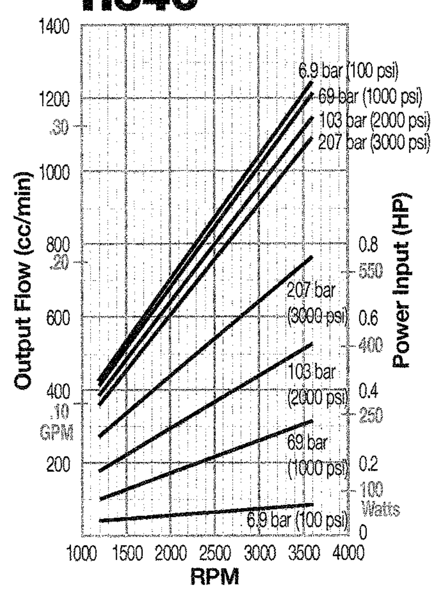
H259



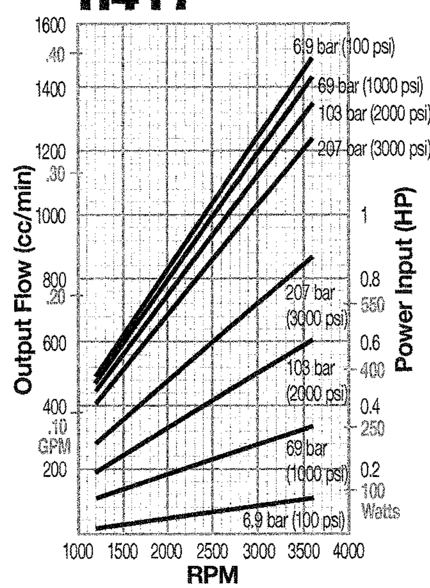
H311



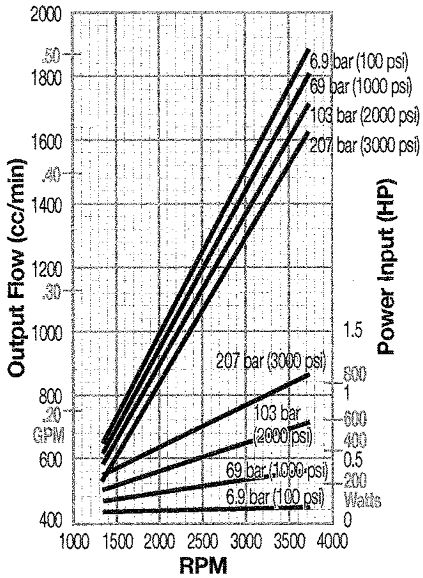
H346



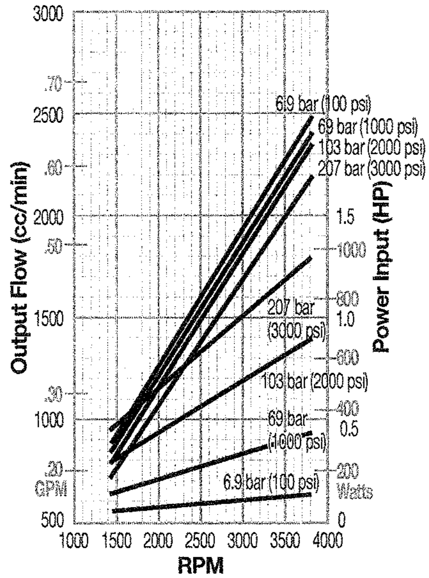
H417



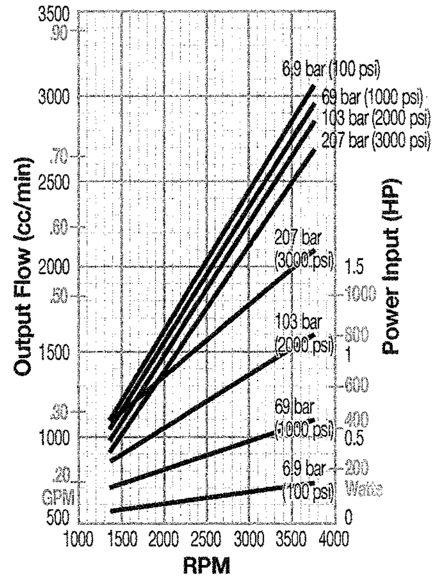
H519



H692



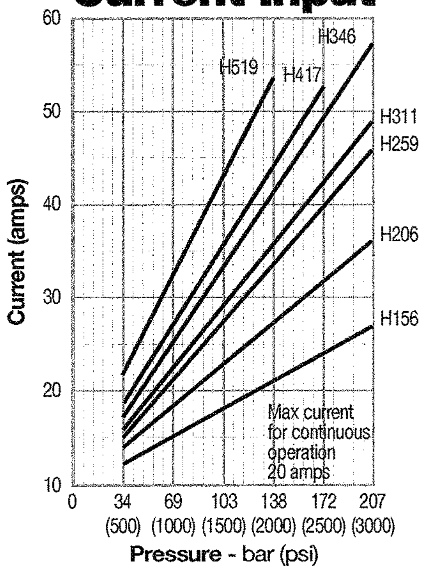
H865



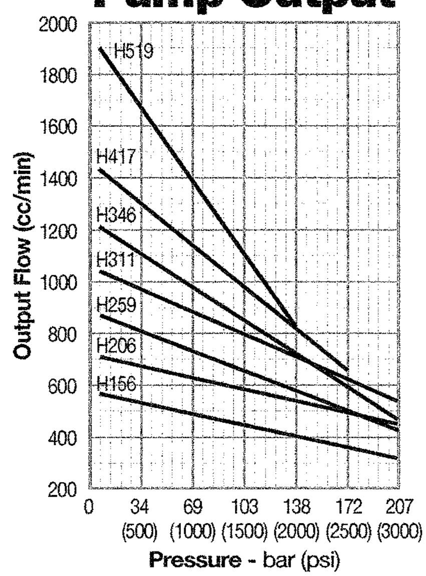
Typical Performance Data

at 12 VDC as assembled with a standard DC motor

Current Input

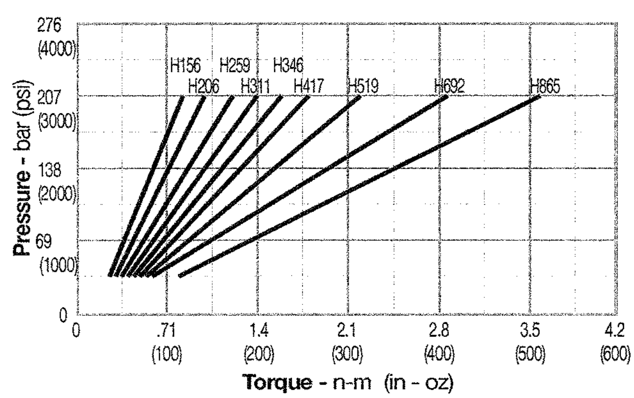


Pump Output



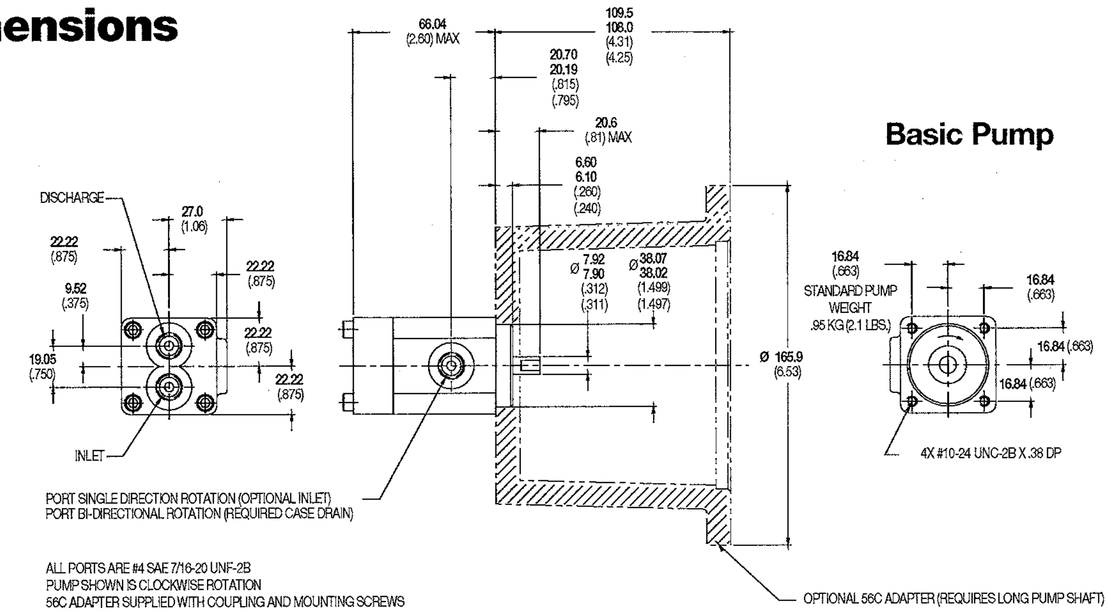
Average Input Torque

Speed: 3000 RPM

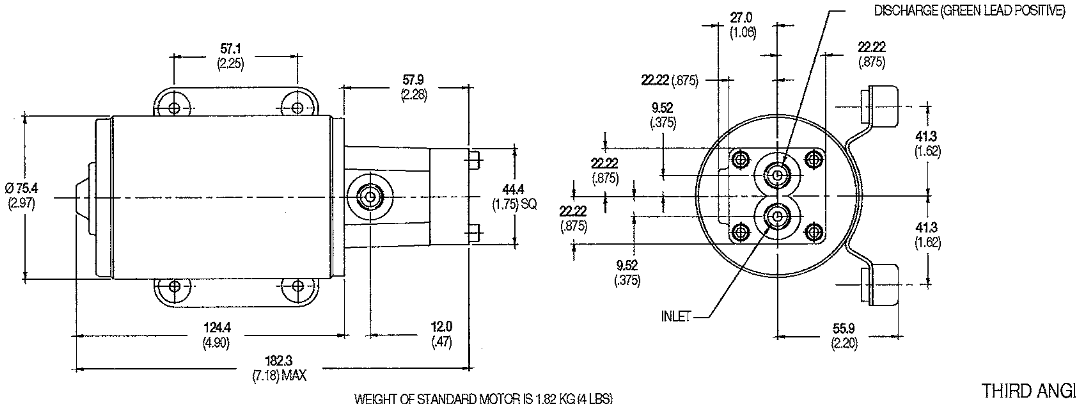


Dimensions

Basic Pump

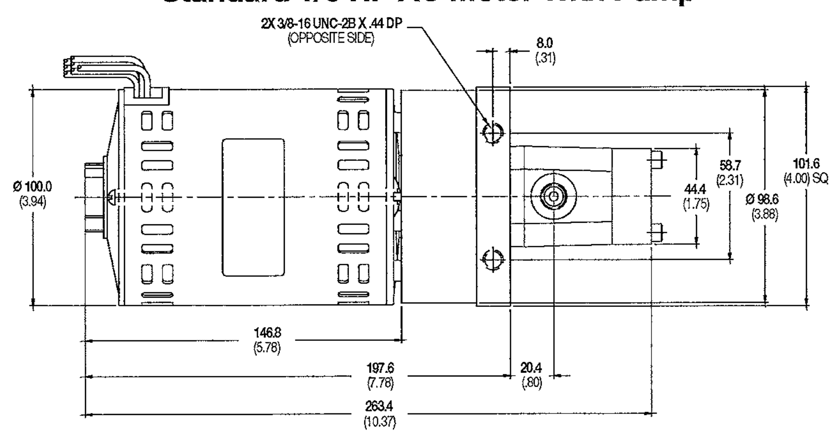


Standard 1/3 HP DC Permanent Magnet Motor With Pump



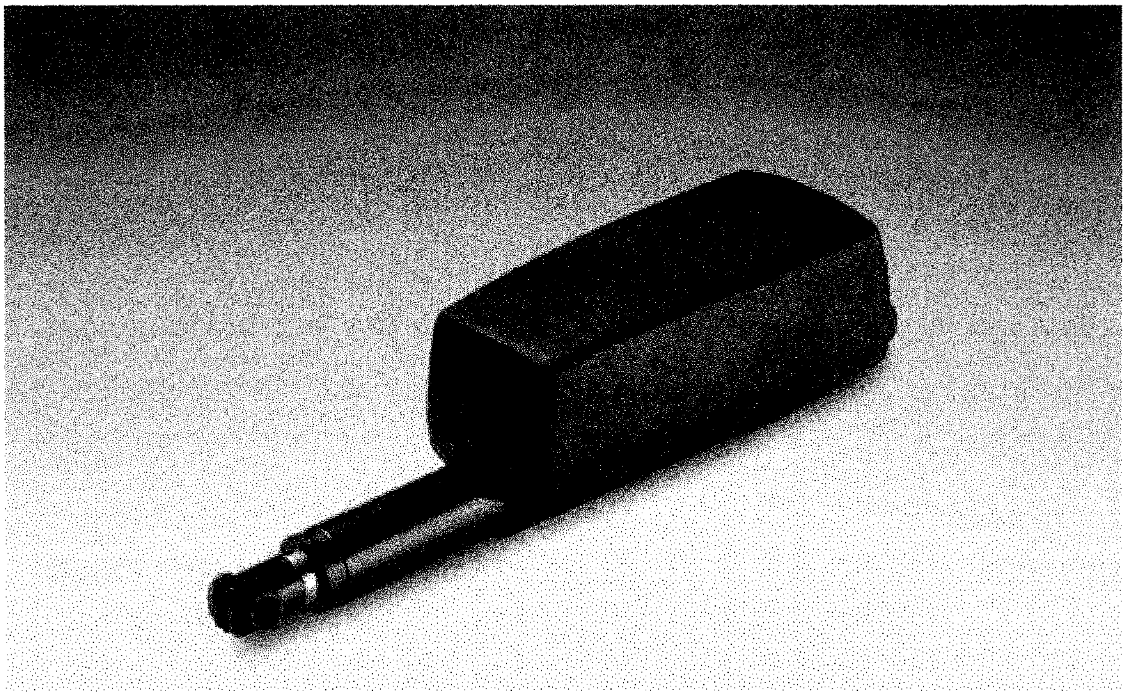
THIRD ANGLE
PROJECTION

Standard 1/3 HP AC Motor With Pump



Note: All dimensions in **mm** (inches).

ACTUATOR LA28



LA28: 12/24 V DC. Max. thrust 3500 N

Features:
- *12/24 V DC permanent magnetic motor*
- *Low noise level*
- *Stainless steel piston rod*
- *Smooth and compact design with small installation dimensions*
- *Re-inforced plastic housing protects motor and gear*
- *2.3 m straight cable with 6.3 mm Jack-plug. On the 12 V version 1.5 m straight cable without plug*
- *Standard protection class: IP 51*
- *Colour: black*
- *Ambient temperature +5° to +40° C*
- *Duty cycle max. 10% or 6 min./hour at continuous use*

Options:
- *Reed-switch*
- *Protection class: IP 65 or IP 66*
- *Brake increase self-locking ability for LA28 actuators with 6 or 9 mm pitch and with or without strong motor*
- *Splines (push only)*
- *Colour: grey*
- *0.2 m and 0.4 m coiled cable*
- *Safety nut for LA28.1/2/3 (push direction only)*
- *For use with LINAK® control boxes, CS16 box only or internal CS card (CS28)*
- *Available with extra powerful motor (S-motor)*

LA28/28S is a very quiet and powerful actuator, designed for use in the furniture, rehabilitation and hospital bed industry. This actuator is also used in agricultural machinery and for a wide range of industrial applications.

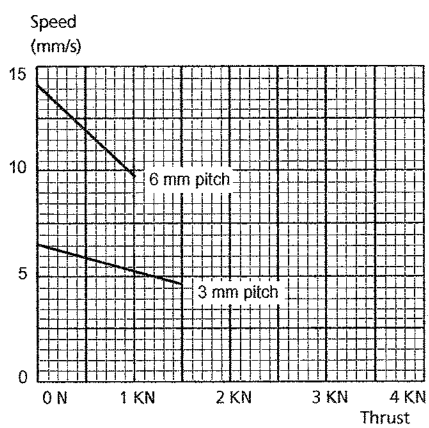
LINAK®

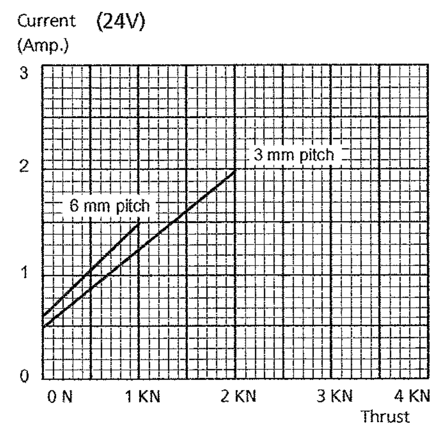
WE IMPROVE YOUR LIFE

Order number	Thrust max. [N]	Self-lock** max. [N]	Typical speed 0/full load [mm/s]	Standard stroke lengths 100 - 400 mm in steps of 50 mm [mm]	Duty cycle [%]	Typical amp.* Full load [A]	
						12 V	24 V
282XXX+XXXXXXXX	1000	500	13 / 9.8	100 400	10		1.5
281XXX+XXXXXXXX	2000	2000	6.5 / 4.1	100 400	10		2
283XXX+4XXXX1XX	1500	1500	32 / 18	100 400	10	8	5
282XXX+4XXXX1XX	2000	2000	22 / 13	100 400	10	8	4.5
285XXX+XXXXX1XX	3000	3000	9 / 5	100 400	10	8.5	4.4
286XXX+XXXXX1XX	3500	3500	7.2 / 4.0	100 400	10	9.0	4.5

* Typical values. Above data: Measurements are made in connection with a stable power supply.

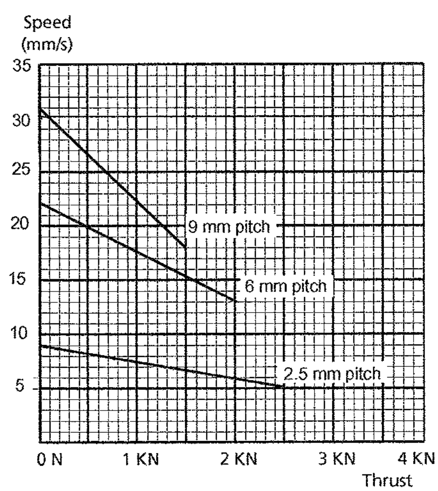
** LINAK control boxes are designed so that they will short-circuit the motor terminals (poles) of the actuator(s) when the actuator(s) are not running. This solution gives the actuator(s) a higher self-locking ability. If the actuator(s) are not connected to a LINAK control box the terminals of the motor must be short-circuited to achieve the above mentioned self-locking ability.

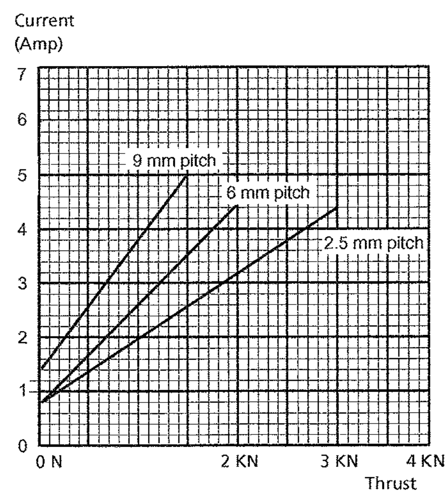




Current Amp./24 V DC.
Current requirement is doubled on the 12 V DC version

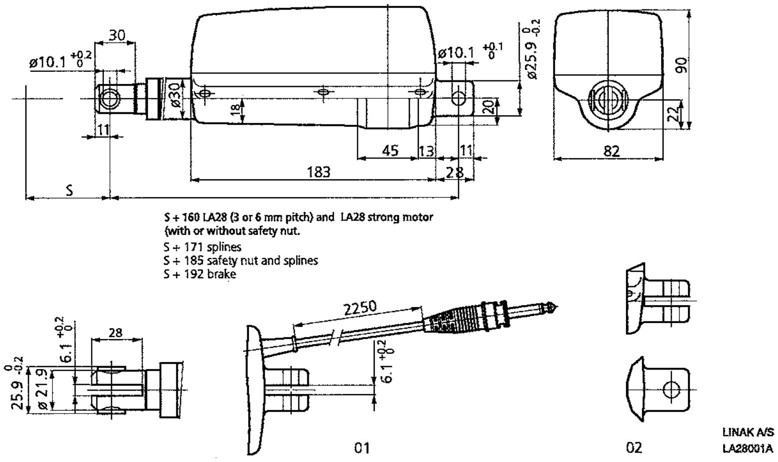
Below curves are for LA28 with S-motor and brake:





Actuators with 2.5 mm pitch only with S-motor

Above data are average figures.
The measurements are made with the actuators connected to a stable power supply.



LA28 and LA28S

Ordering example:

28 1 1 0 0 + 0 0 250 0 0 0

Actuator type

Spindle type:
1 = 3 mm (pitch)
2 = 6 mm (pitch)
3 = 9 mm (pitch)
4 = 4 mm (pitch)
5 = 2.5 mm (pitch)
6 = 2 mm (pitch)
7 = 12 mm (pitch)

Back fixture:
1 = Standard
2 = Turned 90 degrees

Positioning/CS:
0 = Standard
R = Reed-switch
Y = CS28A mount. (Cable without plug 1.0 m). (Only IP 51))
Z = CS28B mount. (Cable without plug 1.0 m and handset cable). Handset cable only black
C = CS28C mount. (5 core cable without plug 1.0 m). (Cable only in black)

Option: 0 = Standard

Colour:
+ = Grey, - = Black

Option:
0 = Standard
1 = Safety nut
2 = Steel splines
3 = Spline with safety nut
4 = Brake
5 = Brake with safety nut

Installation dim.:
0 = Standard
X = Another installation dimension

Stroke length:
xxx = mm (Max. 400 mm, min. 100 mm)

Motor type:
0 = 24 V
1 = 24 V S-motor
2 = 12 V S-motor (Cable without plug Length = 1.5 m)

IP-degree:
0 = IP 51
1 = IP 65
2 = IP 66

Cables:
Jack-cables
0 = Straight 2.3 m
1 = Straight 1.05 m
2 = Coiled 0.4 m
3 = Coiled 0.2 m
Other cables:
4 = Cable without plug 1.5 m
5 = Cable without plug 1.0 m
(for use with LA28 with CS28 A/B/C)

ULTRAMOTION

High precision at an afford

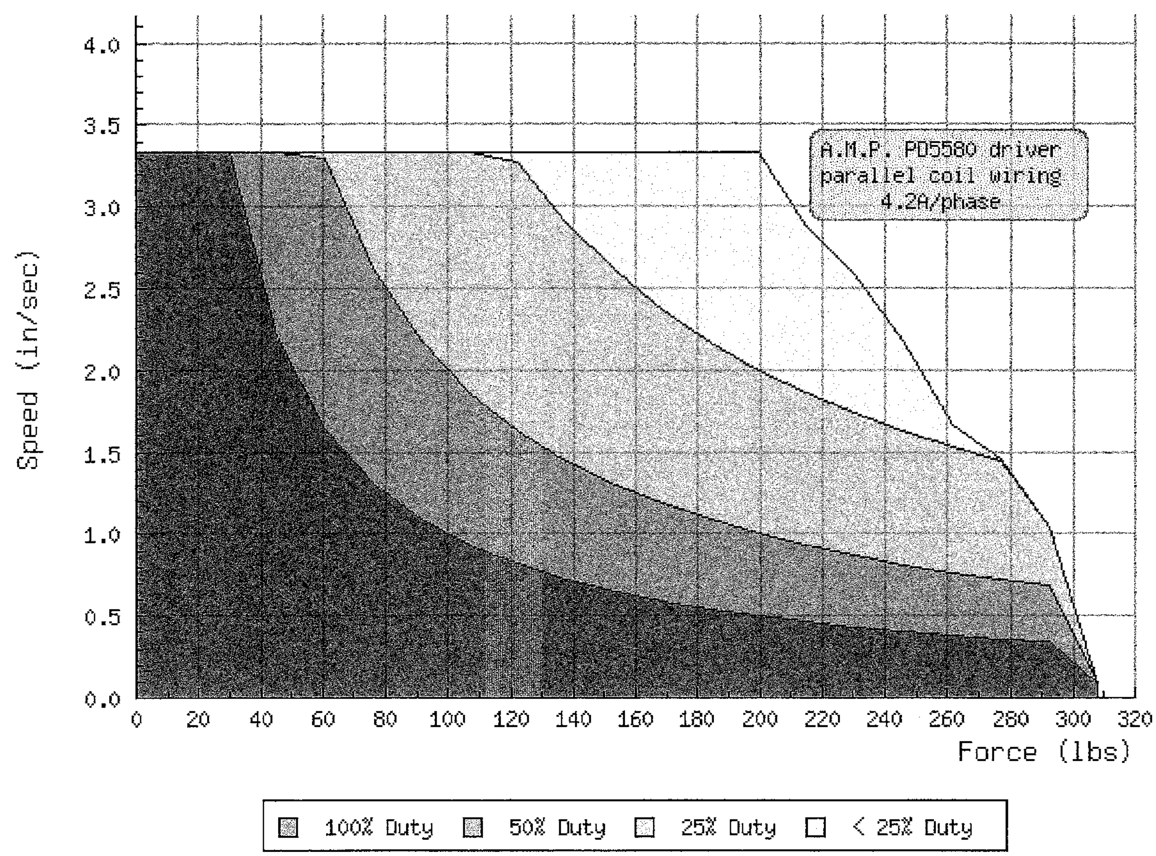
Wednesday 21st of April 2004 08

Printable Version	Ultra Motion **Digit** linear

part number: **D-A.083-HT23-8-P-/4**

max force: 307 lbs
max speed: 3.3 in/sec
backlash: 0.0005 in
linear step resolution: 2400 steps/in

actuator speed/force graph: **D - A.083 - HT23**



lead screw: **A.083** - Acme nut, 0.0833 in/rev lead screw

pitch: 0.0833 in/rev
efficiency: 48 % (self locking)
dynamic load: 100 lb*in/sec (100% duty cycle)

motor: **HT23** - High-torque NEMA 23 size stepper motor

coil impedance: 1 ohms
coil inductance: 1.6 mH
coil current: 3 amps
coil voltage: 3 volts
step angle: 1.8 degrees
drawing: PDF DXF

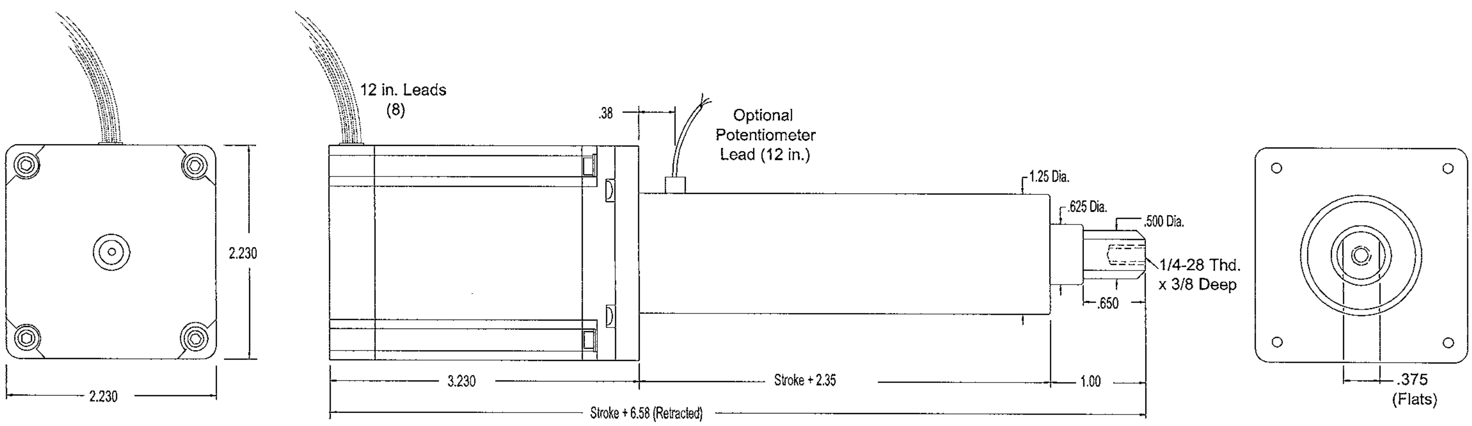
stroke length: **8** inches

potentiometer: **P** - Precision linear potentiometer

total resistance: 10 K ohms
linearity: 2 %

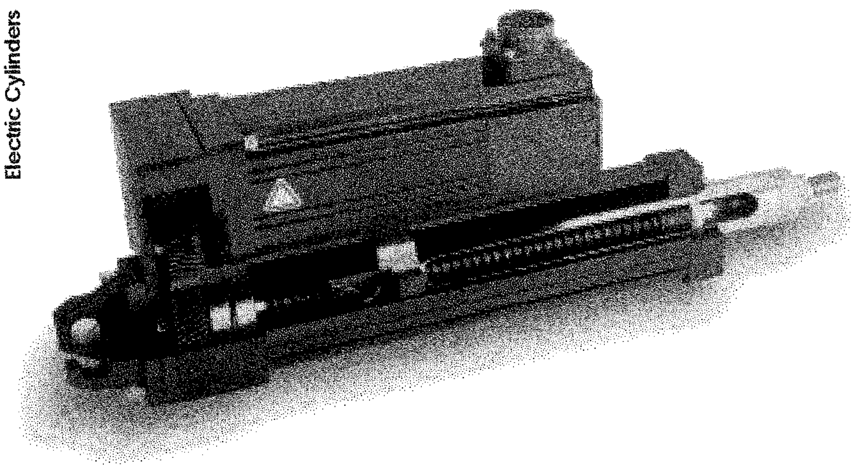
nose mount: **4** - 1/4-28 UNF threaded hole

CALL FOR INFORMATION / 631.298.9179



DIGIT (NEMA 23)

Overview



N2 Series electric cylinders are available with four motor types to meet a variety of application requirements. The N2-D family features a cost effective 24 VDC motor. When combined with D Series controls, the complete system provides simple extend–retract motion, positioning to pre-determined stopping locations, or positioning to an analog voltage command; all at the lowest installed cost.

Operating with a powerful 160 VDC motor, the N2-H family of cylinders are ideally suited for high load and duty cycle applications. Controls provide simple limit switch positioning and edge guiding, or positioning to an analog voltage command.

The N2-S/P family is a step motor based linear actuator. These systems are selected for applications that require high load and duty cycle, in-position holding, open loop operation, repeatable positioning to 0.0005 inches [0.013 mm] and maintenance-free operation.

Industrial Devices' N2-B Series Electric Cylinders offer very high acceleration and duty cycle for the most demanding automated motion applications. The B8000 Servo Drives are designed to optimize the performance of the brushless servo motor.

All N2 Series Cylinders are available with several time-proven options to enhance operation in the industrial environment. Options include holding brakes, linear potentiometers or encoders for position feedback, dual rod-end bearings to increase side load and more. See the end of this section for more information.

		N2-D Series	N2-H Series		N2-S/P Series	N2-B Series
Motor Type		24 VDC Permanent Magnet	160 VDC Permanent Magnet Servo		1.8° Hybrid Stepper	Rare Earth Magnet Brushless Servo
Performance Curves		Page A-160	Page A-166		Page A-172	Page A-182
Load Capacity	lbs [N]	600 [2,670]	600 [2,670]		600 [2,670]	600 [2,670]
No Load Speed	in/s [mm/s]	24 [610]	25 [635]		25 [635]	30 [760]
Repeatability	in [mm]	±0.005 [.127]	± 0.005 [.127]		± 0.0005 [.0127]	± 0.001 [.025]
Compatible Controls Offered		D2200 D2300 D2400 D2500	H3301B H3321B H3501	B8001 B8961 B8962	NextStep SmartStep S6002 S6961 S6962	B8001 B8501 B8961 B8962

Industrial Devices Corporation
707-789-1000 • 800-747-0064 • E-mail: info@idcmotion.com

General Specifications

System Backlash	0.015 inches [0.38 mm]
Thrust Tube	
Side Load Moment	Consult factory.
Rotation	Does not rotate.
Standard Travel Lengths	2, 4, 6, 8, 10, 12, and 16.5 (18–DB); custom stroke lengths available

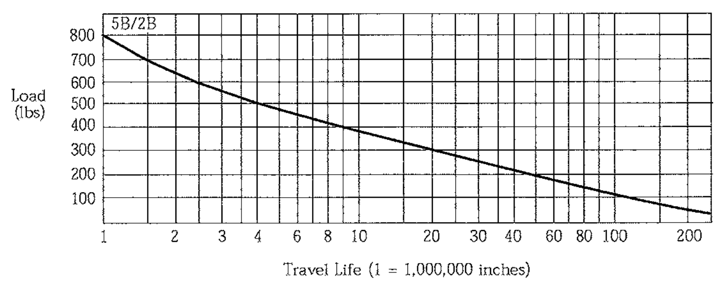
Construction Materials

Bearing Housings	Type 380 die cast aluminum, epoxy coated
Cylinder Housing	6063 T-6 aluminum, hard-coated anodized and Teflon impregnated
Thrust Tube	300 Series stainless steel, 1/8 hard, ground
Wiper Seal	Polyurethane
Lead Screw	
Pitch Choices	2, 5 Ball; 5, 8 Acme
Support Bearings	Ball bearings
Acme Screw; drive nut	0.625 inch diameter, carbon steel screw; lubricated polyacetal plastic (N2-D, N2-P) or bronze (N2-H, N2-S, N2-B) nut
Ball Screw; drive nut	0.625 inch diameter, carbon steel screw; alloy steel, heat-treated ball nut

Life

Acme Screw Life: Usable life for an acme screw is defined as the length of travel completed before backlash (of leadscrew/nut) exceeds 0.020 inches [0.5 mm]. A travel life of 1 million inches under the maximum rated load can be used as a general approximation, however, since wear is directly dependent on application conditions (load, duty cycle, move profiles and environment) it is difficult to quantify an accurate travel life.

Ball Screw Life: Load vs. Travel Life Chart



Weight (Approximate, 2 inch stroke unit without options. Add 0.25 lbs [0.11kg] per additional inch of stroke.)

N2-D Series	7 lbs [3.2 kg]
N2-H Series	9 lbs [4.1 kg]
N2-S/P Series	
N2-P22	6 lbs [2.7 kg]
N2-S32	9 lbs [4.1 kg]
N2-B Series	6 lbs [2.7 kg]

Motor

Specifications/Dimensions	See pages A-194 to A-198

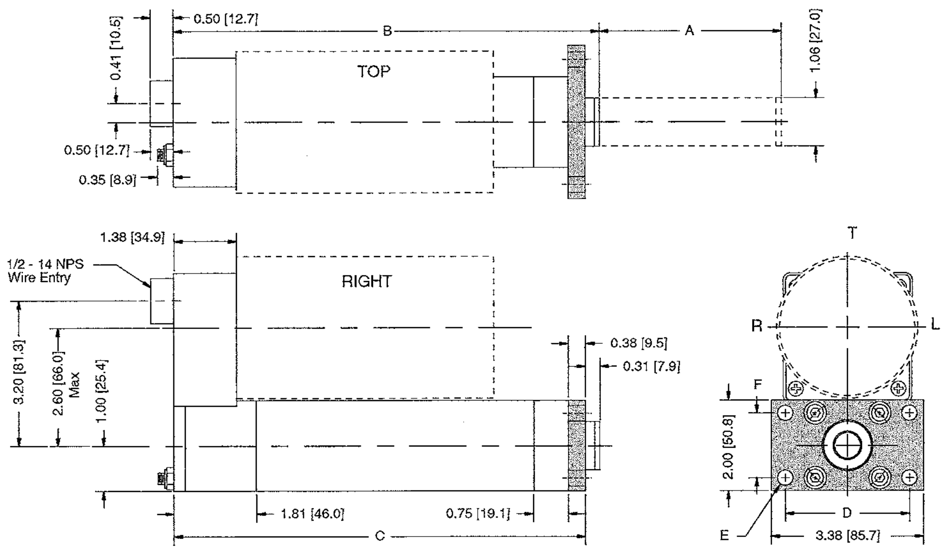
Environmental Operation (See the Options and Accessories section, page A-231.)

Temperature Range	32° to 140°F, [0° to 60°C] -H high temperature option allows 32° to 160°F [0° to 70°C] -F sub-freezing temperature option allows –20° to 105°F [-29° to 40°C]
Moisture	Humid, but not direct moisture contact -W water resistant option allows some direct moisture contact
Contaminants	Non-corrosive, non-abrasive -PB protective boot option prevents moisture and dry contaminants from entering the cylinder through the wiper ring on the rod

Dimensions

MF1 Head Rectangular Flange Mounting
Parallel

- For AutoCAD® DXF drawings, go to our website, or call the factory for a diskette
- For motor dimensions, go to pages A-194 to A-198
- For rod-end dimensions, go to page A-199

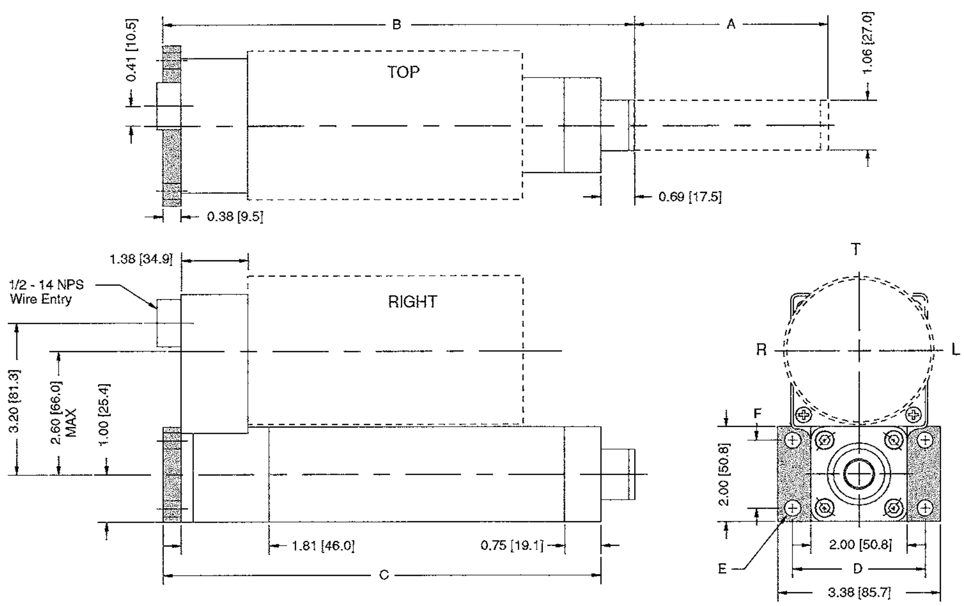


	English Option MF1 (inches)	Metric Option MF1M (mm)
D	2.75	72*
E	0.34	9*
F	1.43	36*

* Meets ISO 40mm bore standard

A Strokes	2.00 (50.8)	8.00 (203.2)	24.00 (609.6)	**B Retract**	stroke + 5.37 (136.4)
	4.00 (101.6)	10.00 (254.0)		**C Mounting**	stroke + 5.06 (128.5)
	6.00 (152.4)	12.00 (304.8)			

MF2 Cap Rectangular Flange Mounting
Parallel



	English Option MF2 (inches)	Metric Option MF2M (mm)
D	2.75	72*
E	0.34	9*
F	1.43	36*

* Meets ISO 40mm bore standard

A Strokes	2.00 (50.8)	8.00 (203.2)	24.00 (609.6)	**B Retract**	stroke + 5.75 (146.1)
	4.00 (101.6)	12.00 (304.8)		**C Mounting**	stroke + 5.06 (128.5)
	6.00 (152.4)	18.00 (457.2)			

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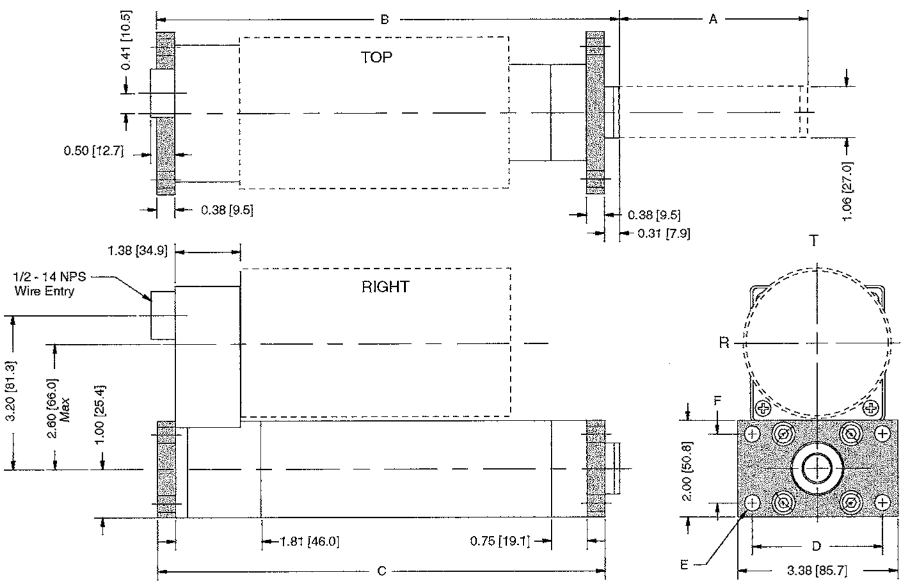
Dimensions

MF3 Rectangular Mounting Flanges
Parallel

- For AutoCAD® DXF drawings, go to our website, or call the factory for a diskette
- For motor dimensions, go to pages A-194 to A-198
- For rod-end dimensions, go to page A-199

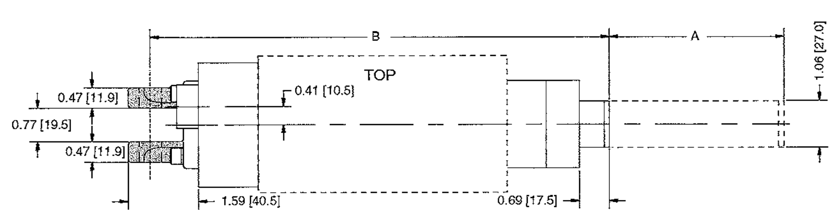


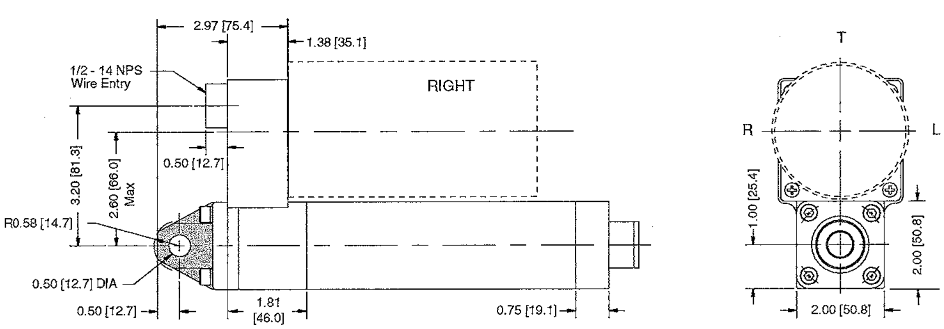
		English Option	Metric Option
		MF3 (inches)	MF3M (mm)
D		2.75	72*
E		0.34	9*
F		1.43	36*

* Meets ISO 40mm bore standard

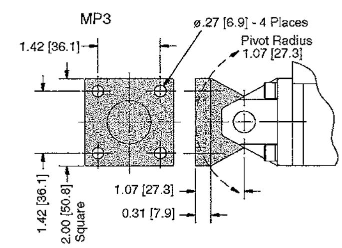
A Strokes	2.00 (50.8)	8.00 (203.2)	24.00 (609.6)	**B Retract**	stroke + 5.75 (146.1)
	4.00 (101.6)	12.00 (304.8)		**C Mounting**	stroke + 5.44 (138.2)
	6.00 (152.4)	18.00 (457.2)			

MP2 Cap Double Clevis Mounting
Parallel





Pivot Base (MP3 only)

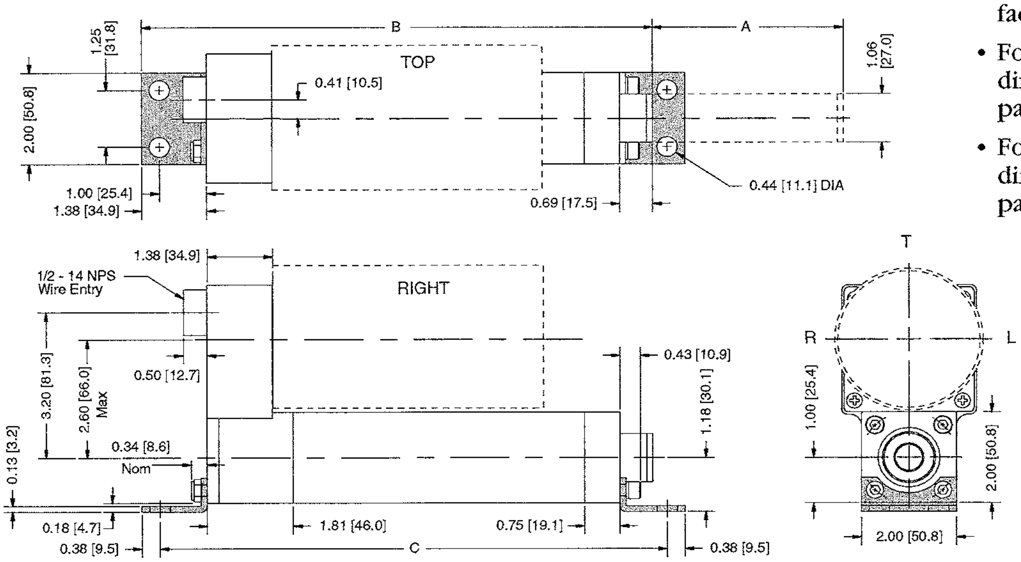


A Strokes	2.00 (50.8)	8.00 (203.2)	24.00 (609.6)	**B Retract**	stroke + 6.47 (164.3)
	4.00 (101.6)	12.00 (304.8)			
	6.00 (152.4)	18.00 (457.2)			

Dimensions

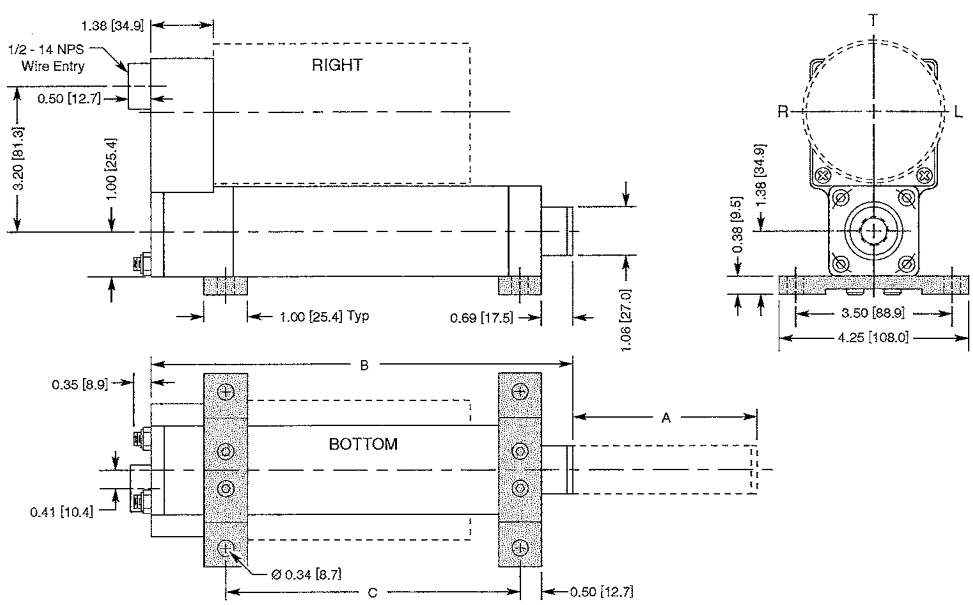
MS1 Side End Angles Mounting
Parallel

- For AutoCAD® DXF drawings, go to our website, or call the factory for a diskette
- For motor dimensions, go to pages A-194 to A-198
- For rod-end dimensions, go to page A-199



A Strokes	2.00	(50.8)	8.00	(203.2)	24.00	(609.6)	**B Retract**	stroke +	6.75	(171.5)
	4.00	(101.6)	12.00	(304.8)			**C Mounting**	stroke +	6.69	(169.9)
	6.00	(152.4)	18.00	(457.2)						

MS2 Side Lugs
Parallel



A Strokes	2.00	(50.8)	8.00	(203.2)	24.00	(609.6)	**B Retract**	stroke +	5.37	(136.4)
	4.00	(101.6)	12.00	(304.8)			**C Mounting**	stroke +	2.56	(65.0)
	6.00	(152.4)	18.00	(457.2)						

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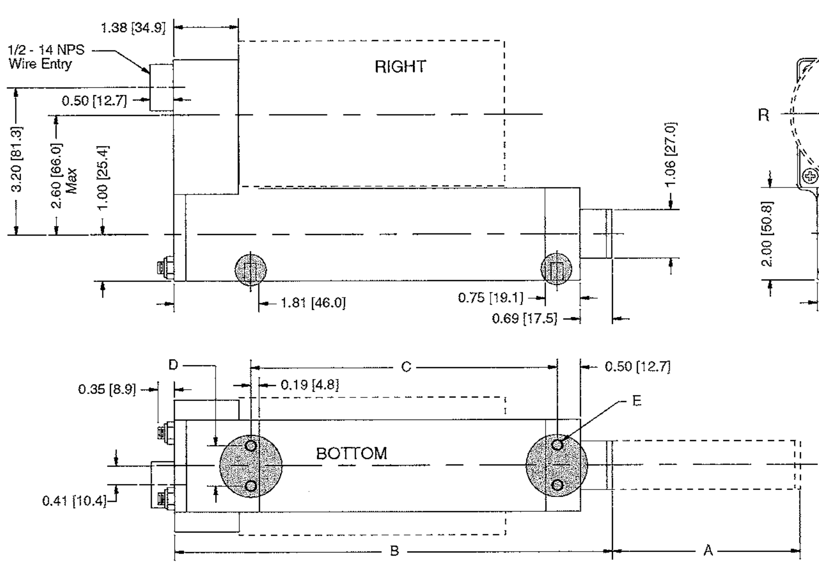
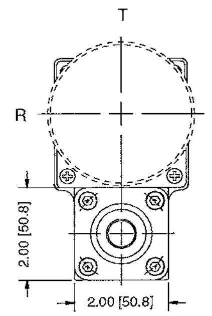
Dimensions

MS6 Side Tapped Mounting

Parallel

- For AutoCAD® DXF drawings, go to our website, or call the factory for a diskette
- For motor dimensions, go to pages A-194 to A-198
- For rod-end dimensions, go to page A-199



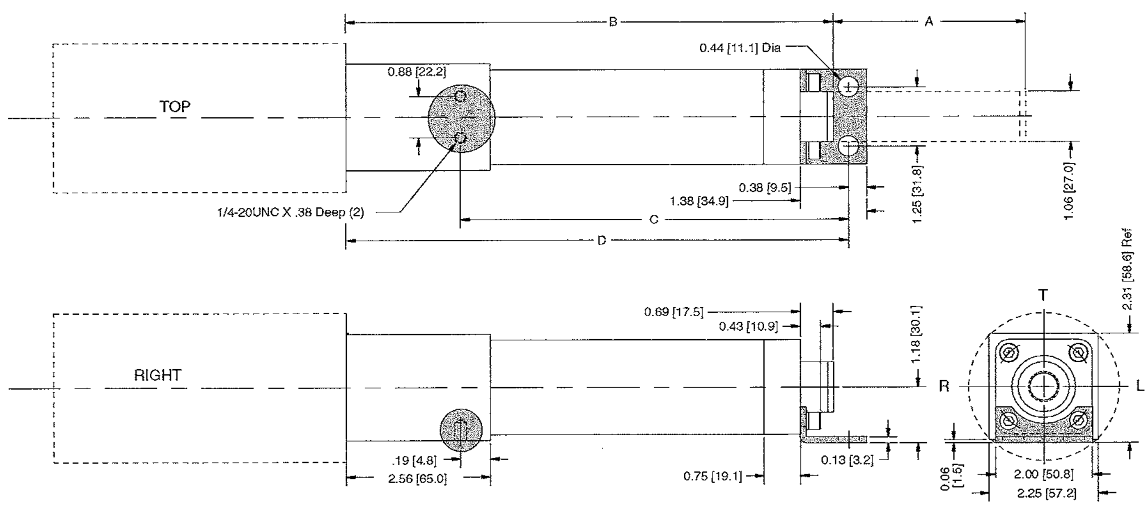


	English Option	Metric Option
	MS6 (inches)	MS6M (mm)
D	0.88	16
E	1/4-20 UNC x 0.38 deep	M6 x 1 x 8 deep

A Strokes	2.00	(50.8)	8.00	(203.2)	24.00	(609.6)	**B Retract**	stroke +	5.37 (136.4)
	4.00	(101.6)	12.00	(304.8)			**C Mounting**	stroke +	2.56 (65.0)
	6.00	(152.4)	18.00	(457.2)					

MS1 Side End Angles Mounting

Inline

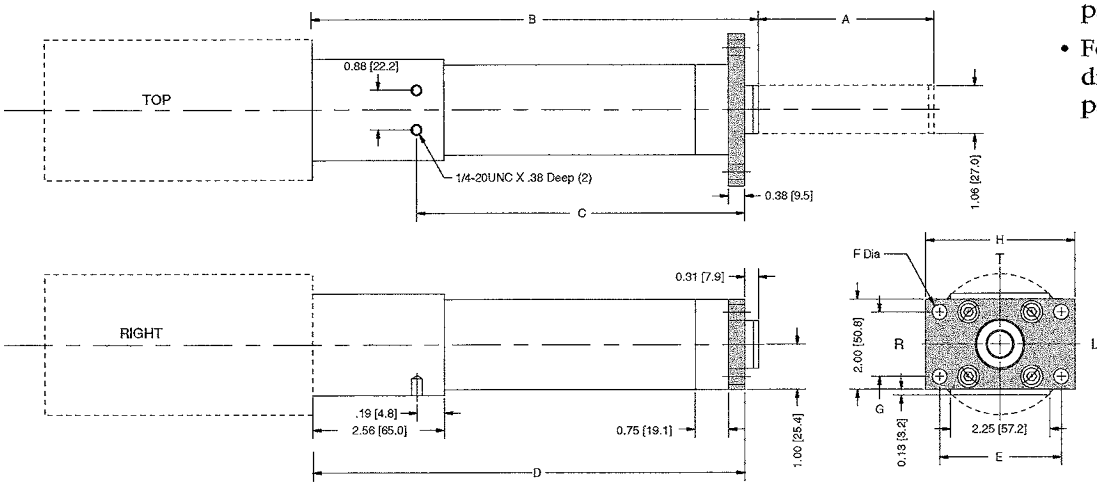


A Strokes	2.00	(50.8)	8.00	(203.2)	24.00	(609.6)	**B Retract**	stroke +	6.12 (155.4)
	4.00	(101.6)	12.00	(304.8)			**C Mounting**	stroke +	4.06 (103.1)
	6.00	(152.4)	18.00	(457.2)			**D Overall**	stroke +	6.43 (163.2)

Dimensions

MF1 Head Rectangular Flange
Inline

- For AutoCAD® DXF drawings, go to our website, or call the factory for a diskette
- For motor dimensions, go to pages A-194 to A-198
- For rod-end dimensions, go to page A-199

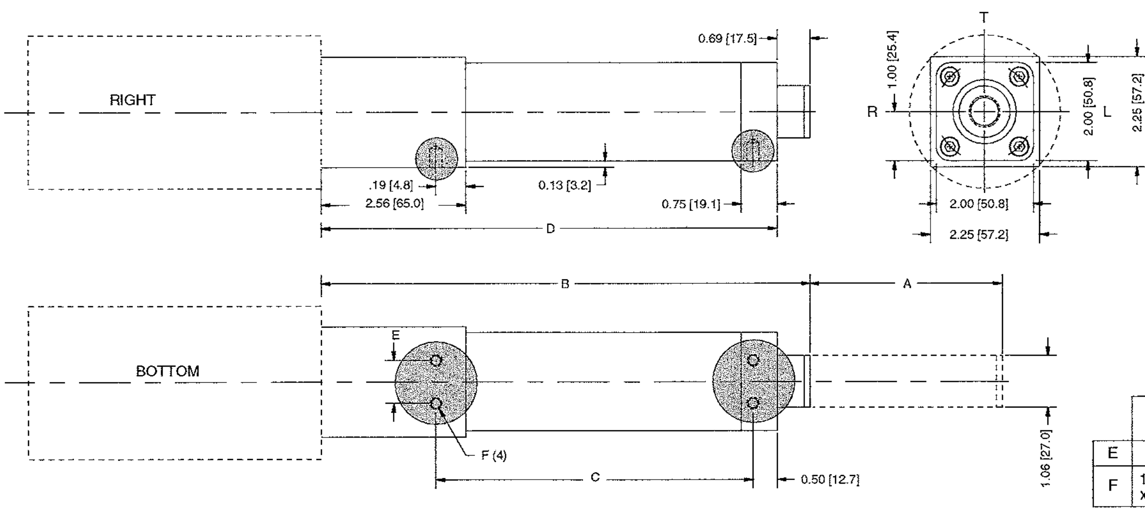


	English Option	Metric Option
	MF1 (inches)	MF1M (mm)
E	2.75	72*
F	0.34	9*
G	1.43	36*
H	3.38	85.7*

* Meets ISO 40mm bore standard

A Strokes	2.00	(50.8)	8.00 (203.2)	24.00 (609.6)	**B Retract**	stroke +	6.12	(155.4)
	4.00	(101.6)	12.00 (304.8)		**C Mounting**	stroke +	3.44	(87.4)
	6.00	(152.4)	18.00 (457.2)		**D Overall**	stroke +	5.81	(147.5)

MS6 Side Tapped Mounting
Inline



	English Option	Metric Option
	MS6 (inches)	MS6M (mm)
E	0.88	16
F	1/4-20 UNC x 0.38 deep	M6 x 1 x 8 deep

A Strokes	2.00	(50.8)	8.00 (203.2)	24.00 (609.6)	**B Retract**	stroke +	6.12	(155.4)
	4.00	(101.6)	12.00 (304.8)		**C Mounting**	stroke +	2.56	(65.0)
	6.00	(152.4)	18.00 (457.2)		**D Overall**	stroke +	5.43	(137.8)

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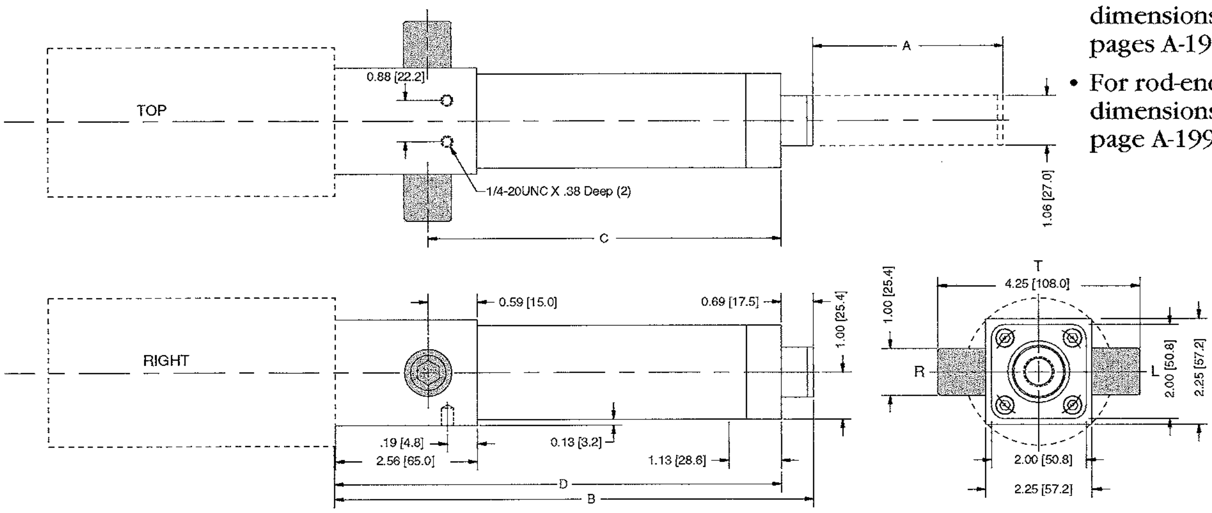
Dimensions

MT4 Trunnion Mounting
Inline

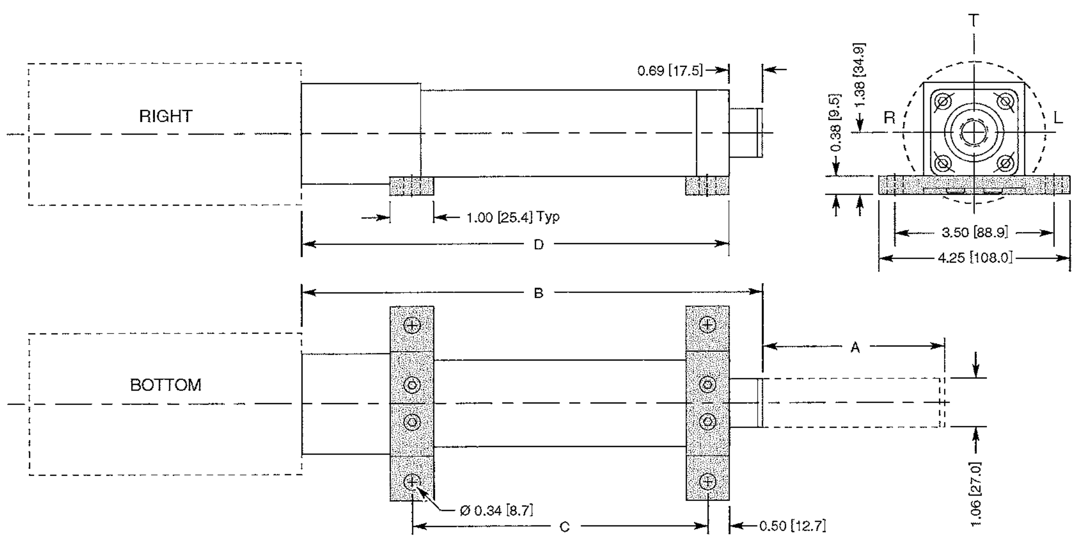
The MT4 mounting replaces the identical MT2 mounting with the same dimensions. The name was changed to be consistent with the EC Series.

- For AutoCAD® DXF drawings, go to our website, or call the factory for a diskette
- For motor dimensions, go to pages A-194 to A-198
- For rod-end dimensions, go to page A-199



A Strokes	2.00	(50.8)	8.00	(203.2)	24.00	(609.6)	**B Retract**	stroke +	6.12	(155.4)
	4.00	(101.6)	12.00	(304.8)			**C Mounting**	stroke +	3.47	(88.1)
	6.00	(152.4)	18.00	(457.2)			**D Overall**	stroke +	5.43	(137.8)

MS2 Side Foot Mounting
Inline



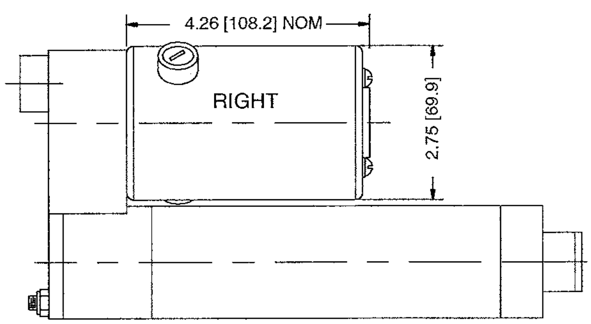
A Strokes	2.00	(50.8)	8.00	(203.2)	24.00	(609.6)	**B Retract**	stroke +	6.12	(155.4)
	4.00	(101.6)	12.00	(304.8)			**C Mounting**	stroke +	2.56	(65.0)
	6.00	(152.4)	18.00	(457.2)			**D Overall**	stroke +	5.43	(137.9)

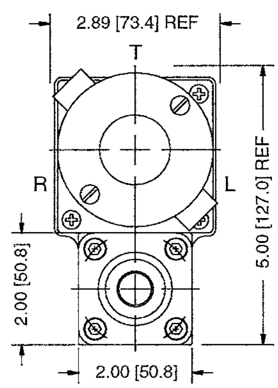
Motor Specifications

N2-D Series	Permanent magnet 2-pole, 24 volt DC motor
Winding Data	D Motor
Inductance	1.8 mH
Resistance	1.0
Torque Constant	8.8 oz-in/Amp
Voltage Constant	6.5 V/krpm
Torque	
Continuous	39.6 oz-in (4.5 Amps)
Peak	88 oz-in (10 Amps)
Rotor Inertia	0.018 oz-in-sec^2
Connections	2 leads, 6 inch [150 mm] length -Q Quick Disconnect option: 3 contact receptacle in anodized or painted aluminum shell, includes 12 ft [3.7 m] cable with molded plug.
Temperature	180°F [82°C] maximum allowable motor case temperature Actual motor case temperature is ambient, duty cycle, speed and load dependent. Refer to speed vs. thrust curves for system duty ratings.

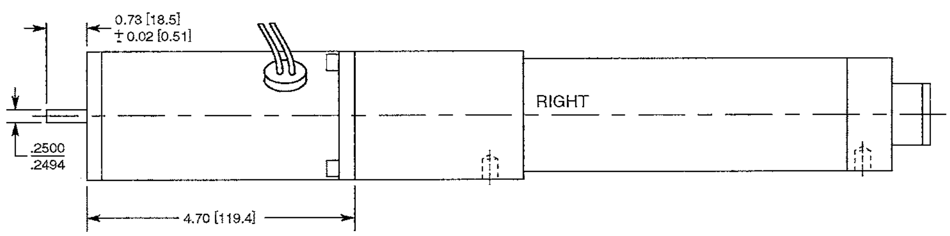
D Motor Dimensions

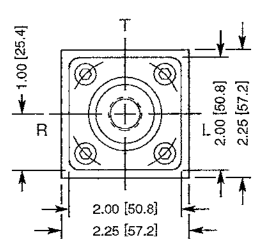
Parallel





Inline





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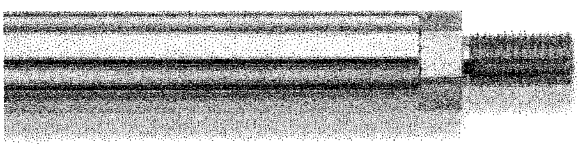
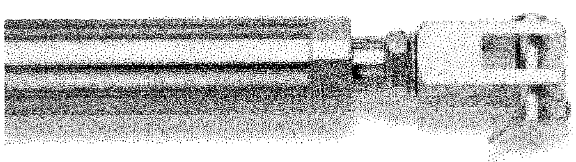
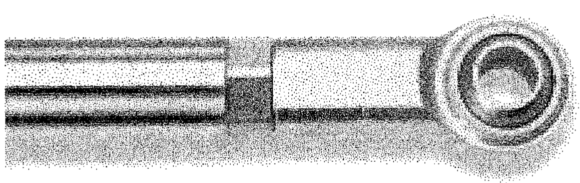
Rod End Dimensions

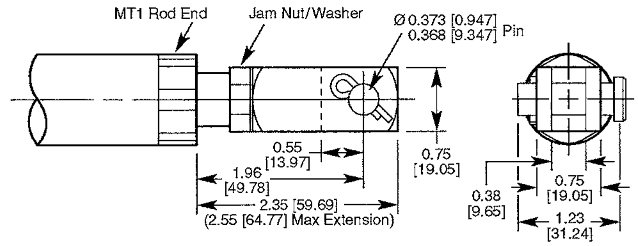




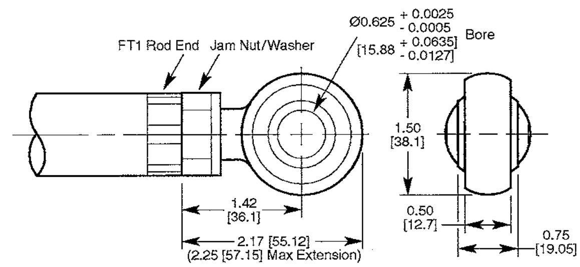


Dimensions in [mm]

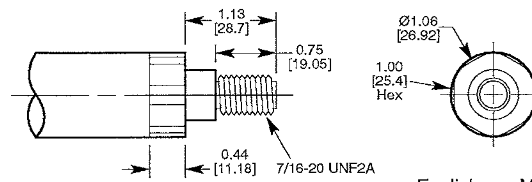
FC2



FS2



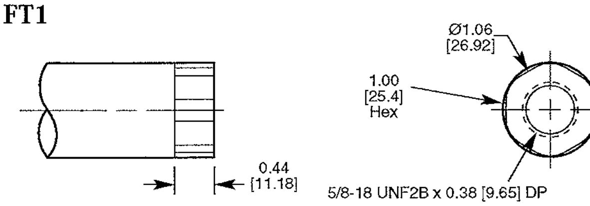
MT1



	English Option	Metric Option
	MT1 (inches)	MT1M (mm)
A	7/16-20 UNF	M12 x 1.25*
B	0.75	24*

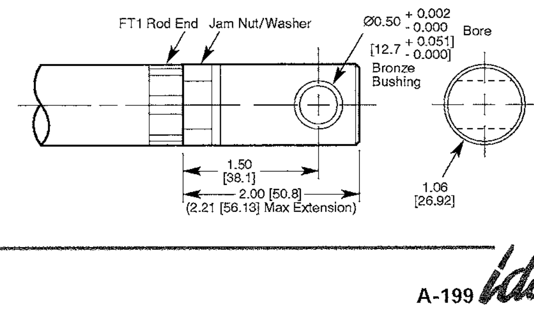
* Meets ISO 40mm bore standard

FT1



	English Option	Metric Option
	FT1 (inches)	FT1M (mm)
A	5/8-18 UNF	M12 x 1.25
B	0.94	24

FE2

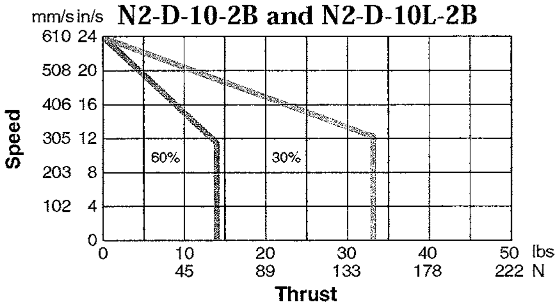


Performance

High-Speed Ballscrew Models

—100% Duty Cycle —60% Duty Cycle —30% Duty Cycle

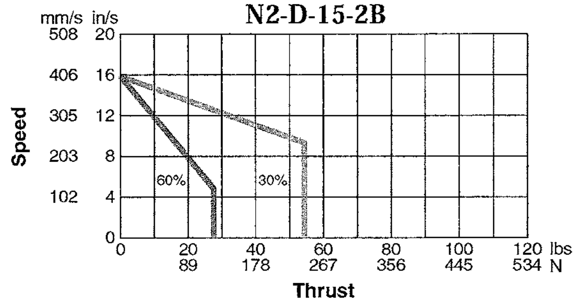
N2-D-10-2B and N2-D-10L-2B



N2-D-10-2B: 1:1 Timing Belt, 2 rev/inch Ballscrew
N2-D-10L-2B: Inline Coupling, 2 rev/inch Ballscrew

Min. Backdrive Load	10 lbs	45 N
Max. No-Load Accel.	180 in/s^2	4572 mm/s^2
Repeatability	±0.010 in	±0.254 mm

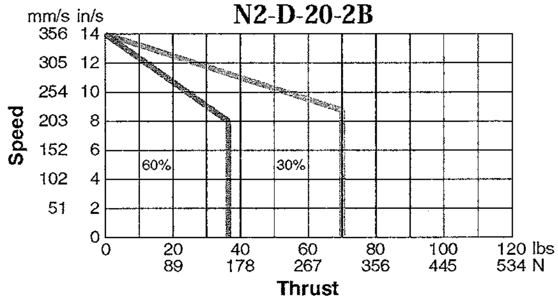
N2-D-15-2B



N2-D-15-2B: 1.5:1 Timing Belt, 2 rev/inch Ballscrew

Min. Backdrive Load	10 lbs	45 N
Max. No-Load Accel.	150 in/s^2	3810 mm/s^2
Repeatability	±0.010 in	±0.254 mm

N2-D-20-2B



N2-D-20-2B: 2:1 Timing Belt, 2 rev/inch Ballscrew

Min. Backdrive Load	10 lbs	45 N
Max. No-Load Accel.	150 in/s^2	3810 mm/s^2
Repeatability	±0.010 in	±0.254 mm

- Performance using D2200 or D2300 Series Controls.
- Duty Cycle is percentage of actuator "on time" or movement over 10 minute interval.
- **For D2500B control, derate thrust by 50%.**
- Repeatability achievable with D2300 control. Reduce cylinder speed prior to final positioning.

- Consider leadscrew **critical speed** and **column load limits** when specifying longer lengths.

2B

30.0	*Critical Speed (in/sec)*
2 thru 18-DB	***Stroke** (in)*
n/a	*Column Load Limit (lb)*

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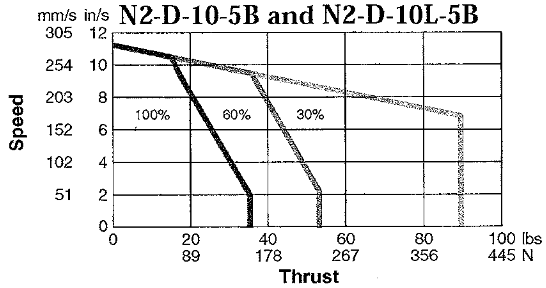
Performance

Ballscrew Models

—100% Duty Cycle —60% Duty Cycle —30% Duty Cycle

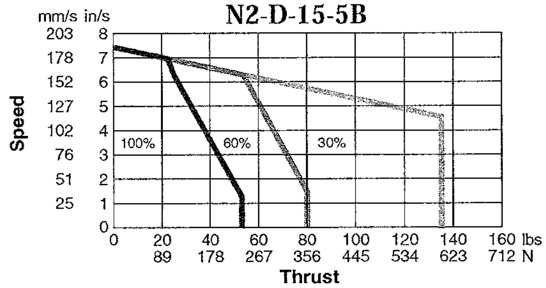
N2-D-10-5B and N2-D-10L-5B



N2-D-10-5B: 1:1 Timing Belt, 5 rev/inch Ballscrew
N2-D-10L-5B: Inline Coupling, 5 rev/inch Ballscrew

Min. Backdrive Load	10 lbs	45 N
Max. No-Load Accel.	180 in/s²	4572 mm/s²
Repeatability	±0.010 in	±0.254 mm

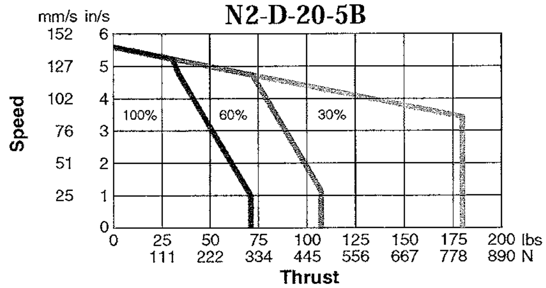
N2-D-15-5B



N2-D-15-5B: 1.5:1 Timing Belt, 5 rev/inch Ballscrew

Min. Backdrive Load	20 lbs	89 N
Max. No-Load Accel.	80 in/s²	2032 mm/s²
Repeatability	±0.005 in	±0.127 mm

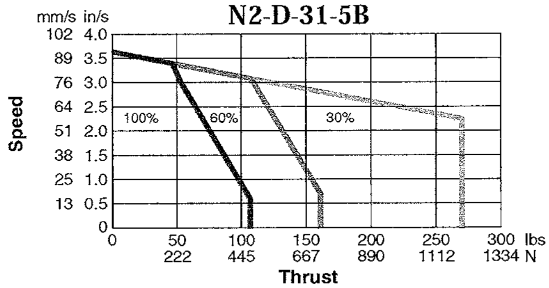
N2-D-20-5B



N2-D-20-5B: 2:1 Timing Belt, 5 rev/inch Ballscrew

Min. Backdrive Load	20 lbs	89 N
Max. No-Load Accel.	70 in/s²	1778 mm/s²
Repeatability	±0.005 in	±0.127 mm

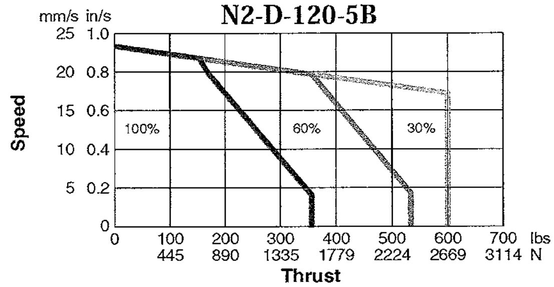
N2-D-31-5B



N2-D-31-5B: 3:1 Helical Gear, 5 rev/inch Ballscrew

Min. Backdrive Load	20 lbs	89 N
Max. No-Load Accel.	40 in/s²	1016 mm/s²
Repeatability	±0.005 in	±0.127 mm

N2-D-120-5B



N2-D-120-5B: 12:1 Helical Gear, 5 rev/inch Ballscrew

Min. Backdrive Load	20 lbs	89 N
Max. No-Load Accel.	13 in/s²	330 mm/s²
Repeatability	±0.005 in	±0.127 mm

- Consider leadscrew **critical speed** and **column load limits** when specifying longer lengths.

5B

15.0	*Critical Speed (in/sec)*
2 thru 18-DB	***Stroke** (in)*
n/a	*Column Load Limit (lb)*

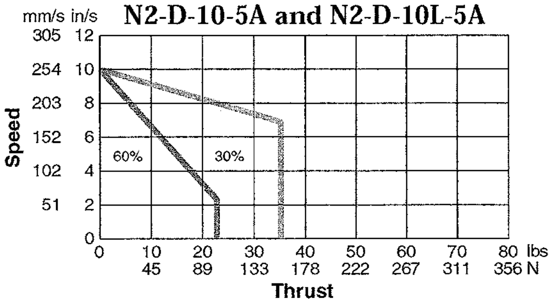
Performance

Electric Cylinders

Acme Screw Models

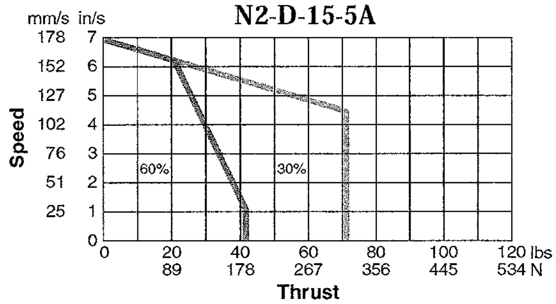
—100% Duty Cycle	—60% Duty Cycle	---30% Duty Cycle

N2-D-10-5A and N2-D-10L-5A



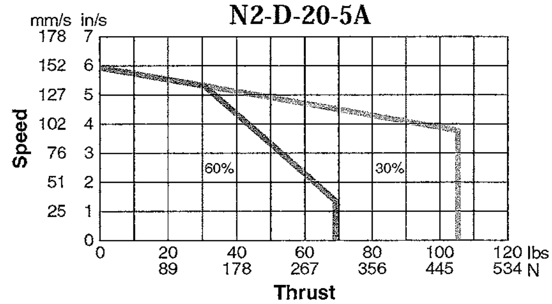
N2-D-10-5A: 1:1 Timing Belt, 5 rev/inch Acme Screw N2-D-10L-5A: 1:1 Inline Coupling, 5 rev/inch Acme Screw		
Min. Backdrive Load	100 lbs	445 N
Max. No-Load Accel.	100 in/s^2	2540 mm/s^2
Repeatability	±0.005 in	±0.127 mm

N2-D-15-5A



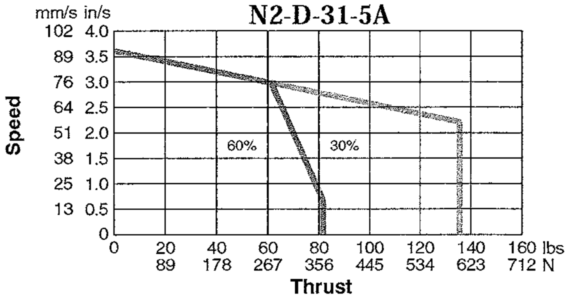
N2-D-15-5A: 1.5:1 Timing Belt, 5 rev/inch Acme Screw		
Min. Backdrive Load	100 lbs	445 N
Max. No-Load Accel.	80 in/s^2	2032 mm/s^2
Repeatability	±0.005 in	±0.127 mm

N2-D-20-5A



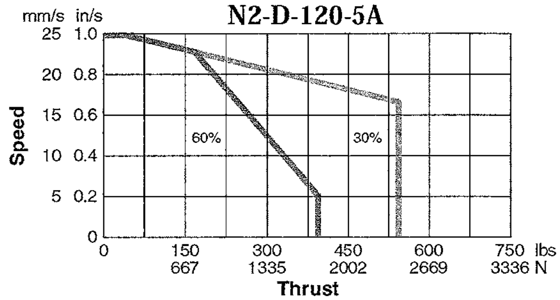
N2-D-20-5A: 2:1 Timing Belt, 5 rev/inch Acme Screw		
Min. Backdrive Load	100 lbs	445 N
Max. No-Load Accel.	70 in/s^2	1778 mm/s^2
Repeatability	±0.005 in	±0.127 mm

N2-D-31-5A



N2-D-31-5A: 3.1:1 Helical Gear, 5 rev/inch Acme Screw		
Min. Backdrive Load	100 lbs	445 N
Max. No-Load Accel.	40 in/s^2	1016 mm/s^2
Repeatability	±0.005 in	±0.127 mm

N2-D-120-5A



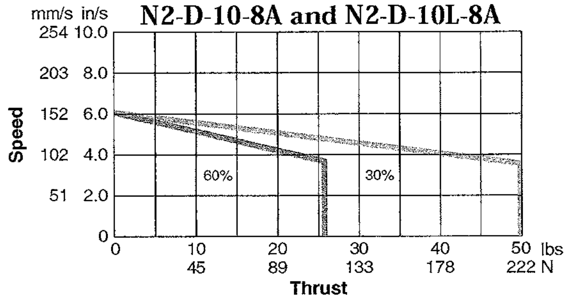
N2-D-120-5A: 12:1 Helical Gear, 5 rev/inch Acme Screw		
Min. Backdrive Load	100 lbs	445 N
Max. No-Load Accel.	13 in/s^2	330 mm/s^2
Repeatability	±0.005 in	±0.127 mm

- Performance using D2200 or D2300 Series Controls.
- Duty Cycle is percentage of actuator "on time" or movement over 10 minute interval.
- **For D2500B control, derate thrust by 50%.**
- Repeatability achievable with D2300 control. Reduce cylinder speed prior to final positioning.

Acme Screw Models

—100% Duty Cycle —60% Duty Cycle —30% Duty Cycle

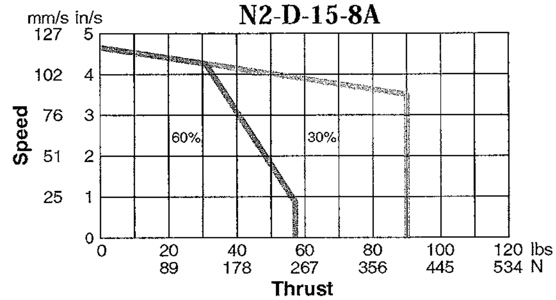
N2-D-10-8A and N2-D-10L-8A



N2-D-10-8A: 1:1 Timing Belt, 8 rev/inch Acme Screw
N2-D-10L-8A: 1:1 Inline Coupling, 8 rev/inch Acme Screw

Min. Backdrive Load	600 lbs	2669 N
Max. No-Load Accel.	60 in/s²	1524 mm/s²
Repeatability	±0.005 in	±0.127 mm

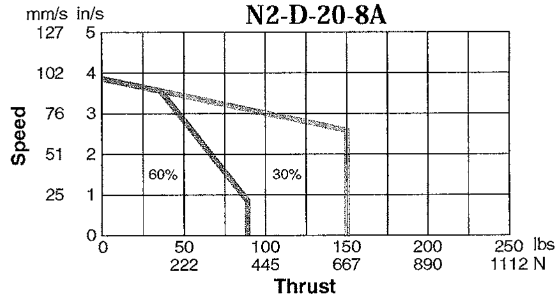
N2-D-15-8A



N2-D-15-8A: 1.5:1 Timing Belt, 8 rev/inch Acme Screw

Min. Backdrive Load	600 lbs	2669 N
Max. No-Load Accel.	50 in/s²	1270 mm/s²
Repeatability	±0.005 in	±0.127 mm

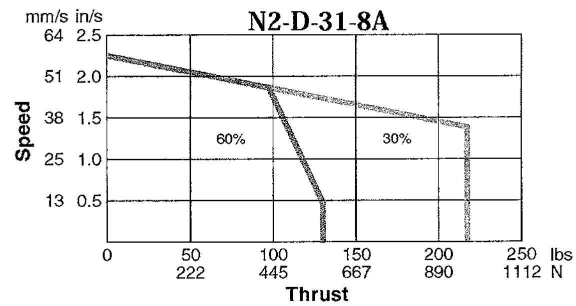
N2-D-20-8A



N2-D-20-8A: 2:1 Timing Belt, 8 rev/inch Acme Screw

Min. Backdrive Load	600 lbs	2669 N
Max. No-Load Accel.	40 in/s²	1016 mm/s²
Repeatability	±0.005 in	±0.127 mm

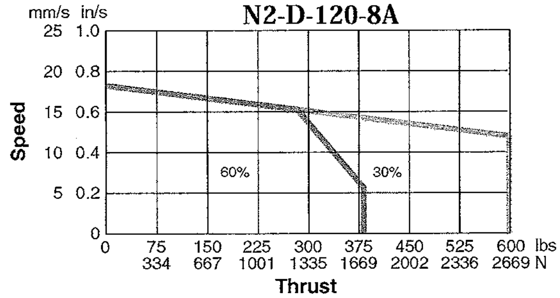
N2-D-31-8A



N2-D-31-8A: 3.1:1 Helical Gear, 8 rev/inch Acme Screw

Min. Backdrive Load	600 lbs	2669 N
Max. No-Load Accel.	25 in/s²	635 mm/s²
Repeatability	±0.005 in	±0.127 mm

N2-D-120-8A



N2-D-120-8A: 12:1 Helical Gear, 8 rev/inch Acme Screw

Min. Backdrive Load	600 lbs	2669 N
Max. No-Load Accel.	8 in/s²	203 mm/s²
Repeatability	±0.005 in	±0.127 mm

5A

15.0	13.8	*Critical Speed (in/sec)*
2 thru 12	18-DB	***Stroke** (in)*
n/a	n/a	*Column Load Limit (lb)*

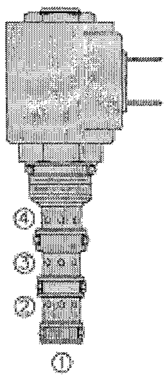
8A

9.4	*Critical Speed (in/sec)*
2 thru 18-DB	***Stroke** (in)*
n/a	*Column Load Limit (lb)*

- Consider leadscrew **critical speed** and **column load limits** when specifying longer lengths.

HydraForce
SV08-40 1.400.1

Solenoid Valves
Spool-Type, 4-Way, 2-Position



Description

A solenoid-operated, 4-way, 2-position, direct-acting spool-type, screw-in hydraulic cartridge valve.

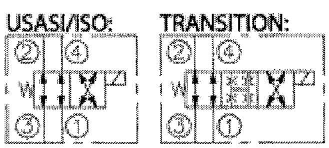
Operation

When de-energized, the cartridge's flow paths are (3) to (2), and (4) to (1).
When energized, the cartridge's spool shifts to open (3) to (4), and (2) to (1). All ports are open at cross-over.
Operation of Manual Override Feature: To override, push button in and twist counterclockwise 180°. The internal spring will push the button out. In this position, the valve may be only partially shifted.
To assure full override shift, pull the button out to its fullest extension and hold it in this position.
To return to normal operation, push button in, twist clockwise 180°, and release. Override will be detented in this position.

Features

- Continuous-duty rated solenoid.
- Hardened precision spool and cage for long life.
- Optional coil voltage and terminations.
- Efficient wet-armature construction.
- All ports may be fully pressurized.
- Cartridges are voltage interchangeable.
- Manual override option.
- Optional waterproof E-coils rated up to IP69K
- Unitized, molded coil design.
- Compact size.

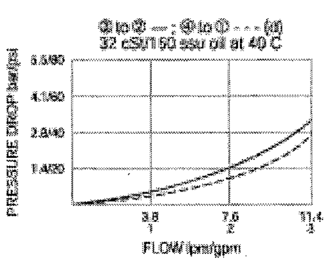
Symbols



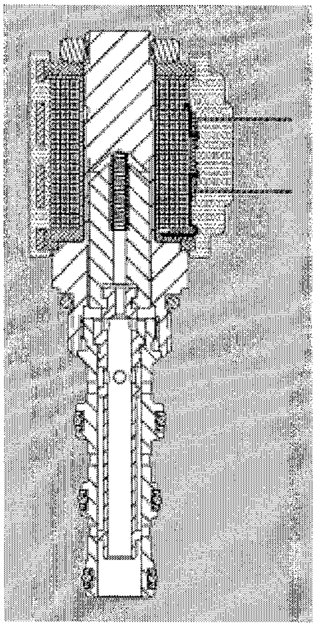
Ratings

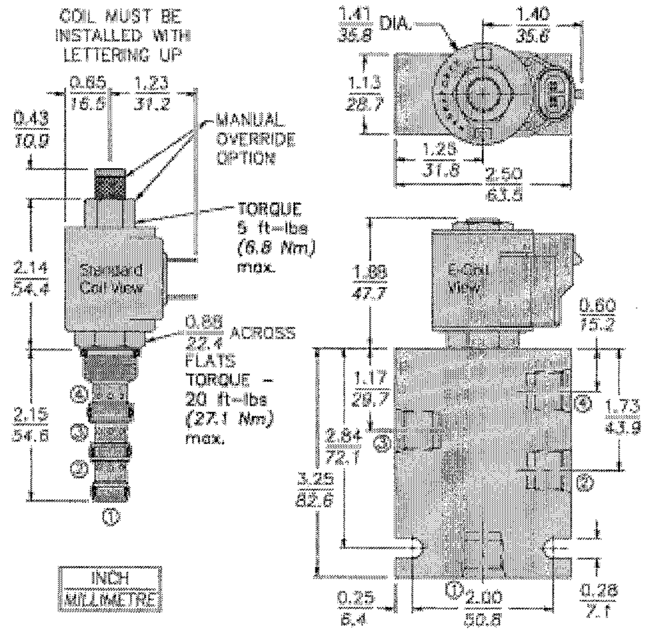
Operating Pressure: 207 bar (3000 psi)
Flow: See Performance Chart
Internal Leakage: 82 cc/minute (5 cu. in./minute) max. at 207 bar (3000 psi)
Temperature: -40 to 120° C with standard Buna seals
Coil Duty Rating: Continuous from 85% to 115% of nominal voltage
Initial Coil Current Draw at 20° C: Standard Coil: 1.2 amps at 12 VDC; 0.13 amps at 115 VAC (full wave rectified); E-Coil: 1.4 amps at 12 VDC; 0.7 amps at 24 VDC
Minimum Pull-In Voltage: 85% of nominal at 207 bar (3000 psi)
Filtration: See page 9.010.1
Fluids: Mineral-based or synthetics with lubricating properties at viscosities of 7.4 to 420 cSt (50 to 2000 ssu)
Installation: No restrictions See page 9.020.1
Cavity: VC08-4 See page 9.108.1
Cavity Tool: CT08-4XX See page 8.600.1
Seal Kit: SK08-4X-MMM See page 8.650.1
Coil Nut: Part No. 7004400; For E-coils manufactured prior to 01-01-04 See page 3.400.1

Performance



Dimensions





Materials

Cartridge: Weight: 0.13 kg. (0.28 lbs.): Steel with hardened work surfaces. Zinc-plated exposed surfaces; Buna N O-rings and polyester elastomer back-up standard.

Standard Ported Body: Weight: 0.27 kg. (0.60 lbs.); Anodized high-strength 6061 T6 aluminum alloy, rated to 240 bar (3500 psi). Ductile iron and steel bodies available; consult factory. See page 8.008.1

Standard Coil: Weight: 0.11 kg. (0.25 lbs.); Unitized thermoplastic encapsulated, Class H high temperature magnetwire. See page 3.200.1

E-Coil: Weight: 0.14kg. (0.3 lbs.); Fully encapsulated with rugged external metal shell; Rated up to IP69K with integral connectors; NOTE: For all E-Coil retrofit applications see page 3.400.1

To Order

SV08-40__ - __ __ __ - __ - __ __ __ __ __

Option
None (Blank)
Manual Override **M**

Porting
Cartridge Only **0**
SAE 6 **6T**
1/4 in. BSP* **2B**
3/8 in. BSP* **3B**
*BSP Body: U.K. Mfr. Only

Seals
Buna N (Std.) **N**
Fluorocarbon **V**

Voltage Std. Coil
0 Less Coil**
10 10 VDC†
12 12 VDC
24 24 VDC
36 36 VDC
48 48 VDC
24 24 VAC
115 115 VAC
230 230 VAC
**Includes Std. Coil Nut
† DS, DW or DL terminations only

E-Coil
10 10 VDC
12 12 VDC
20 20 VDC
24 24 VDC

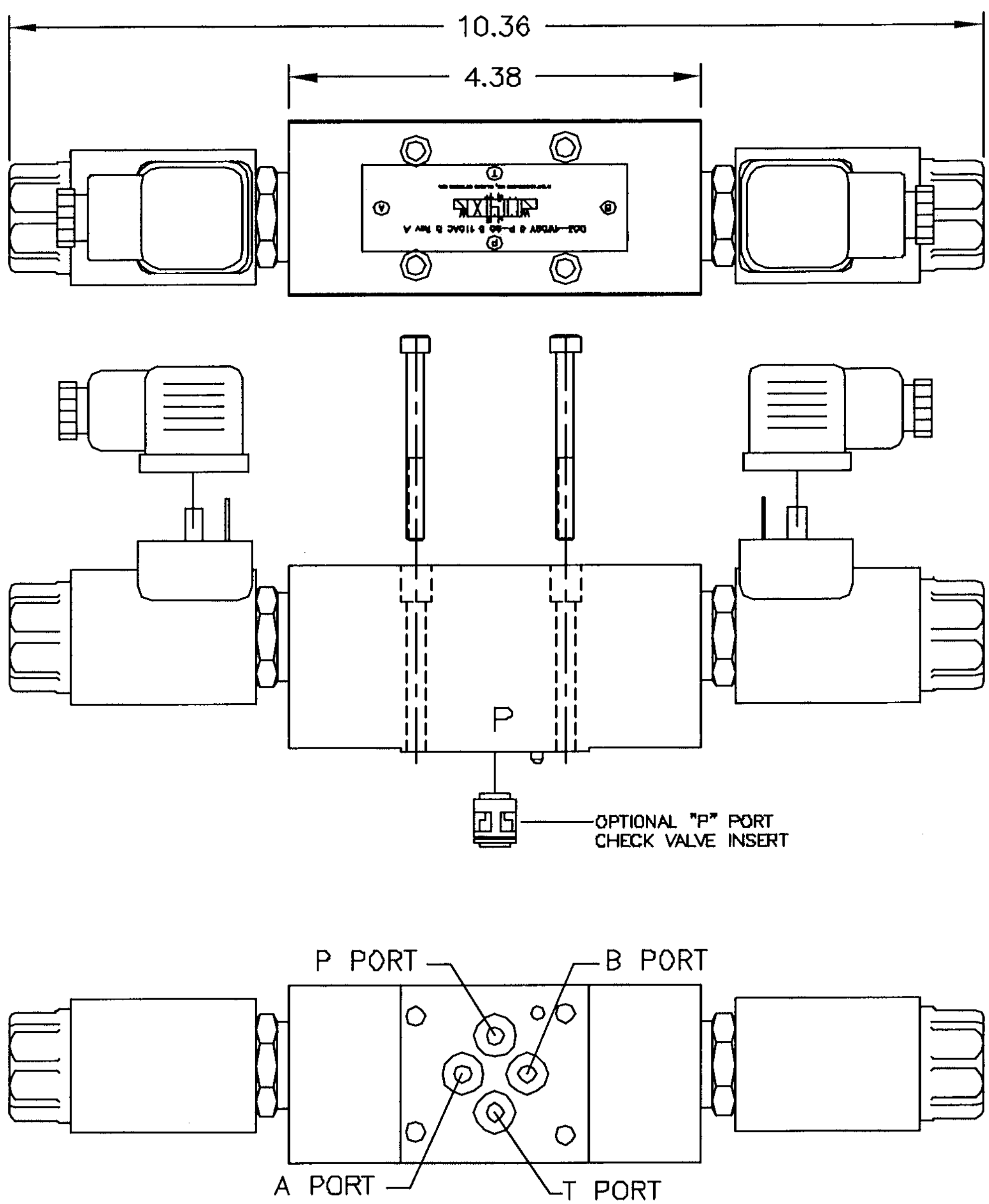
Termination (VDC) Std. Coil
DS Dual Spades
DG DIN 43650
DL Leadwires (2)
DL/W Leads w/Weatherpak® Connectors
DR Deutsch DT04-2P

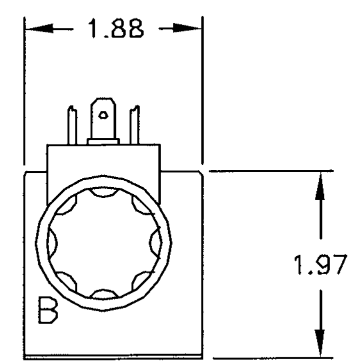
Termination (VAC) Std. Coil
AG DIN 43650
AP 1/2 in. Conduit

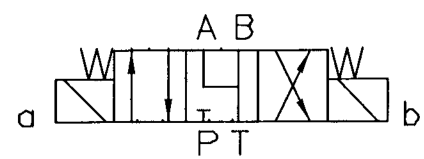
Termination (VDC) E-Coil
ER Deutsch DT04-2P (IP69K Rated)
EY Metri-Pack® 150 (IP69K Rated)

Coils with internal diode are available. Consult factory.

REV	DESCRIPTION	DATE





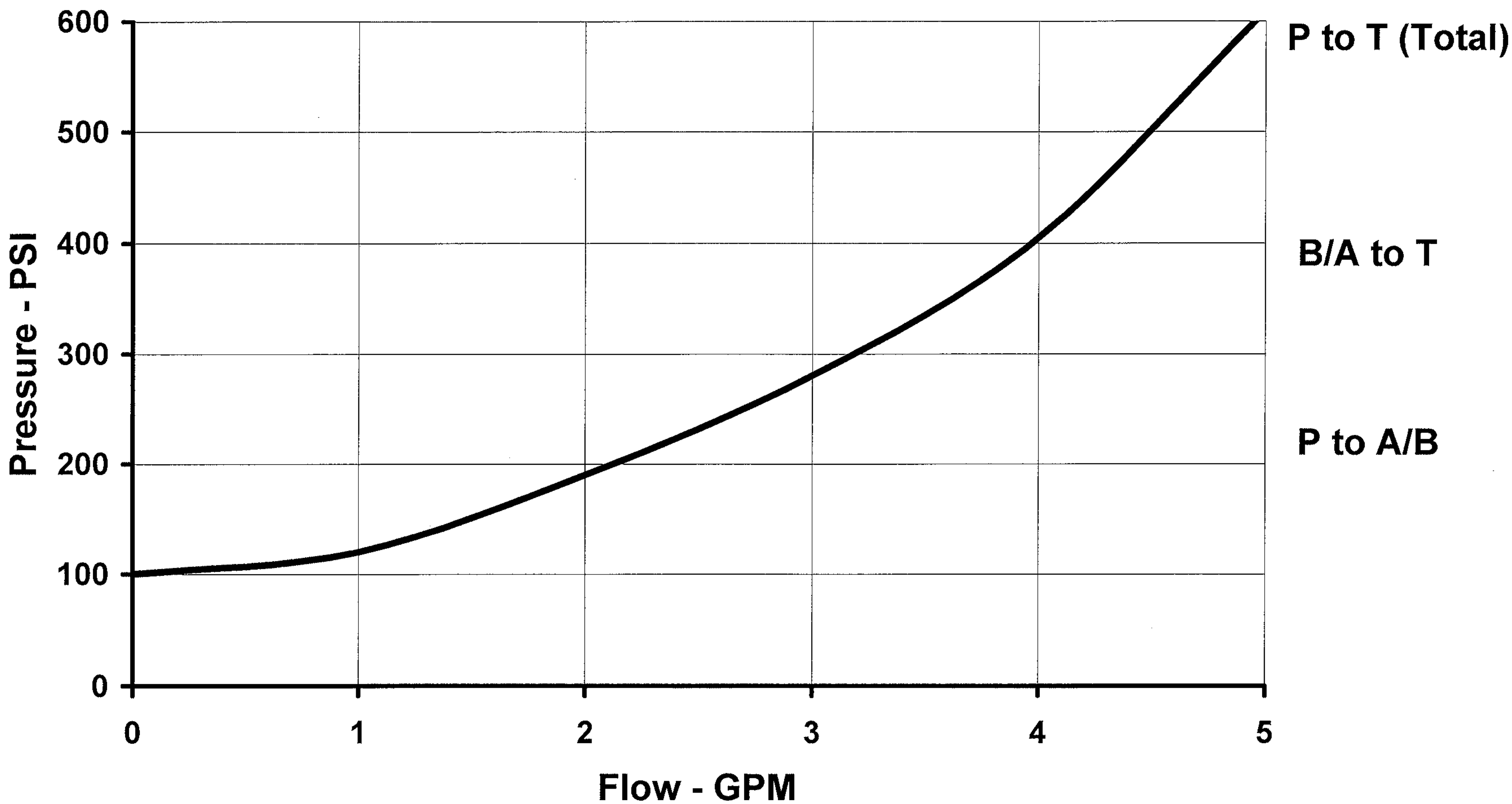


USASI SYMBOL

INTERFACE DEVICES, INC.
230 DEPOT ROAD
MILFORD, CT 06460

SCALE: HALF	TITLE: GENERAL ARRANGEMENT, 6K SOL POPPET 4-WAY VALVE
DWN BY: WNI	
DATE: 2/17/98	REF:
SHT: 1 OF: 1	DWG NO: V03-0001 REV: —

Flow vs. Pressure Drop
IDI Model DO3 4WOSV (with P Port Check Valve)
Shell Tellus 32 @ 70 Deg F



GENERAL PURPOSE 5 OR 10 VOLT OUTPUT PRESSURE SENSORS

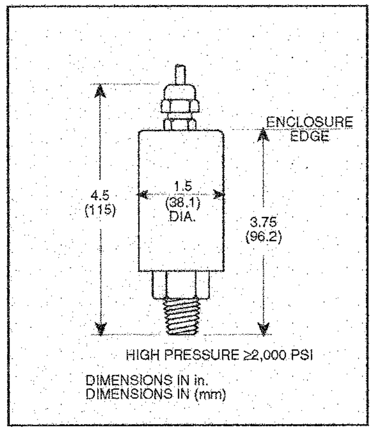
PX303 Series
15 to 10,000 psi

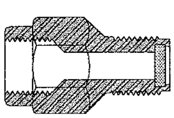
3 YEAR WARRANTY · MADE IN USA · CE

PX303 Series
$255

Connections

BLACK	Com.
RED	EX+
WHITE	SIG+





¼ NPT Pressure Snubbers $10
PS-4G = Gas
PS-4E = Lt Oil
PS-4D = Dense Lq

SPECIFICATIONS
5V Output (10V Output)
Excitation: 9 to 30 Vdc (10 to 30 Vdc) unregulated
Output: 0.5-5.5 (1-11) Vdc
Accuracy: 0.25% FS (linearity, hysteresis, repeatability)
Zero Balance: ± 0.4% FS (±0.2%)
Span Tolerance: ±0.8% FS (±0.4%)
Long Term Stability: ±0.5% FS
Typical Life: 100 million cycles
Operating Temperature: 0 to 160°F (-18 to 71°C)
Compensated Temperature: 30 to 130 °F (-1 to 54°C)
Total Thermal Effects: 1% FS max
Proof Pressure: 200%, 13000 PSI max
Quiescent Exc.: 15 mA maximum
Min Load Resistance: 2000Ω
Response Time: 1 msec
Gage Type: Stainless steel diaphragm, silicone oil filled semiconductor sensor
Shock: 50 g @ 11msec
Vibration: 15 g 10-2000 Hz
Wetted Parts: 17-4 PH and 300 Series Stainless Steel
Pressure Port: ¼ NPT male
Press. Cavity: 0.075 cubic inches
Electrical Conn.: 3 cond, 22 AWG, PVC unshielded, 3 ft (1 m) cable
Weight: 7.8 oz (221 g) to 1000 psi 9.9 oz (281 g) from 1000 psi
Snubbers protect sensors from fluid spikes/hammers!

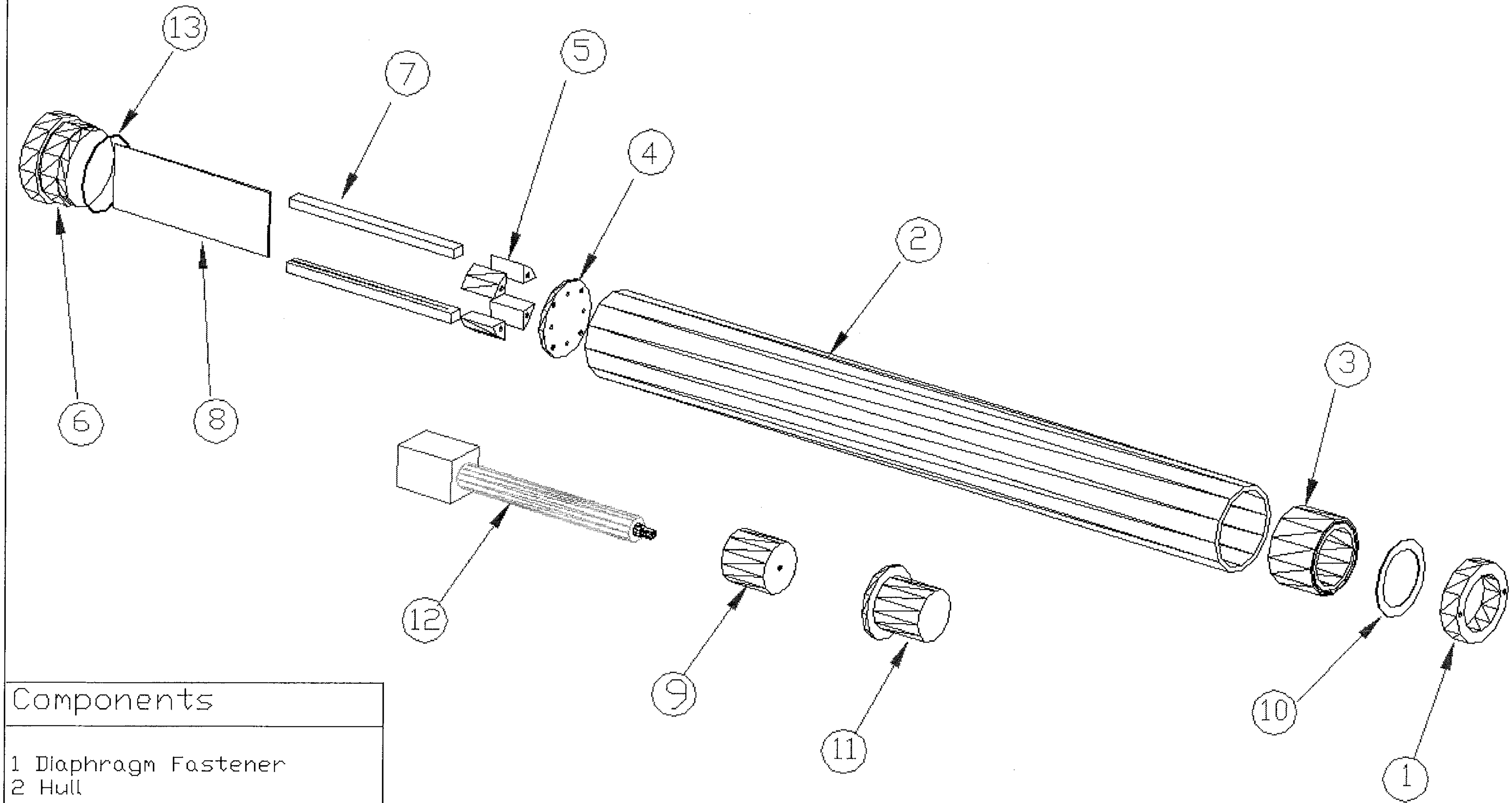
MOST POPULAR MODELS HIGHLIGHTED

To Order ***(Specify Model Number)***

RANGE	5 VOLT OUTPUT MODELS	PRICE	COMPATIBLE METERS
GAGE MODELS			
0 to 15 psig	**PX303-015G5V**	**$255**	DP41-E, DP25-E, DP460-E
0 to 50 psig	**PX303-050G5V**	**255**	DP41-E, DP25-E, DP460-E
0 to 100 psig	**PX303-100G5V**	**255**	DP41-E, DP25-E, DP460-E
0 to 200 psig	**PX303-200G5V**	**255**	DP41-E, DP25-E, DP460-E
0 to 300 psig	**PX303-300G5V**	**255**	DP41-E, DP25-E, DP460-E
0 to 500 psig	**PX303-500G5V**	**255**	DP41-E, DP25-E, DP460-E
0 to 1000 psig	**PX303-1KG5V**	**255**	DP41-E, DP25-E, DP460-E
0 to 2000 psig	**PX303-2KG5V**	**255**	DP41-E, DP25-E, DP460-E
0 to 3000 psig	**PX303-3KG5V**	**255**	DP41-E, DP25-E, DP460-E
0 to 5000 psig	**PX303-5KG5V**	**255**	DP41-E, DP25-E, DP460-
0 to 7500 psig	**PX303-7.5KG5V**	**255**	DP41-E, DP25-E, DP460-E
0 to 10000 psig	**PX303-10KG5V**	**255**	DP41-E, DP25-E, DP460-E
ABSOLUTE MODELS			
0 to 15 psia	**PX303-015A5V**	**$255**	DP41-E, DP25-E, DP460-E
0 to 50 psia	**PX303-050A5V**	**255**	DP41-E, DP25-E, DP460-E
0 to 100 psia	**PX303-100A5V**	**255**	DP41-E, DP25-E, DP460-E
0 to 200 psia	**PX303-200A5V**	**255**	DP41-E, DP25-E, DP460-E
0 to 300 psia	**PX303-300A5V**	**255**	DP41-E, DP25-E, DP460-E

To order models with 1-11 Vdc output replace 5V with 10V in model number.
Comes with complete operator's manual.
Ordering Example: PX303-050G10V *is a 1-11 Vdc output pressure sensor,* ***$255.***

APPENDIX C: DRAWINGS

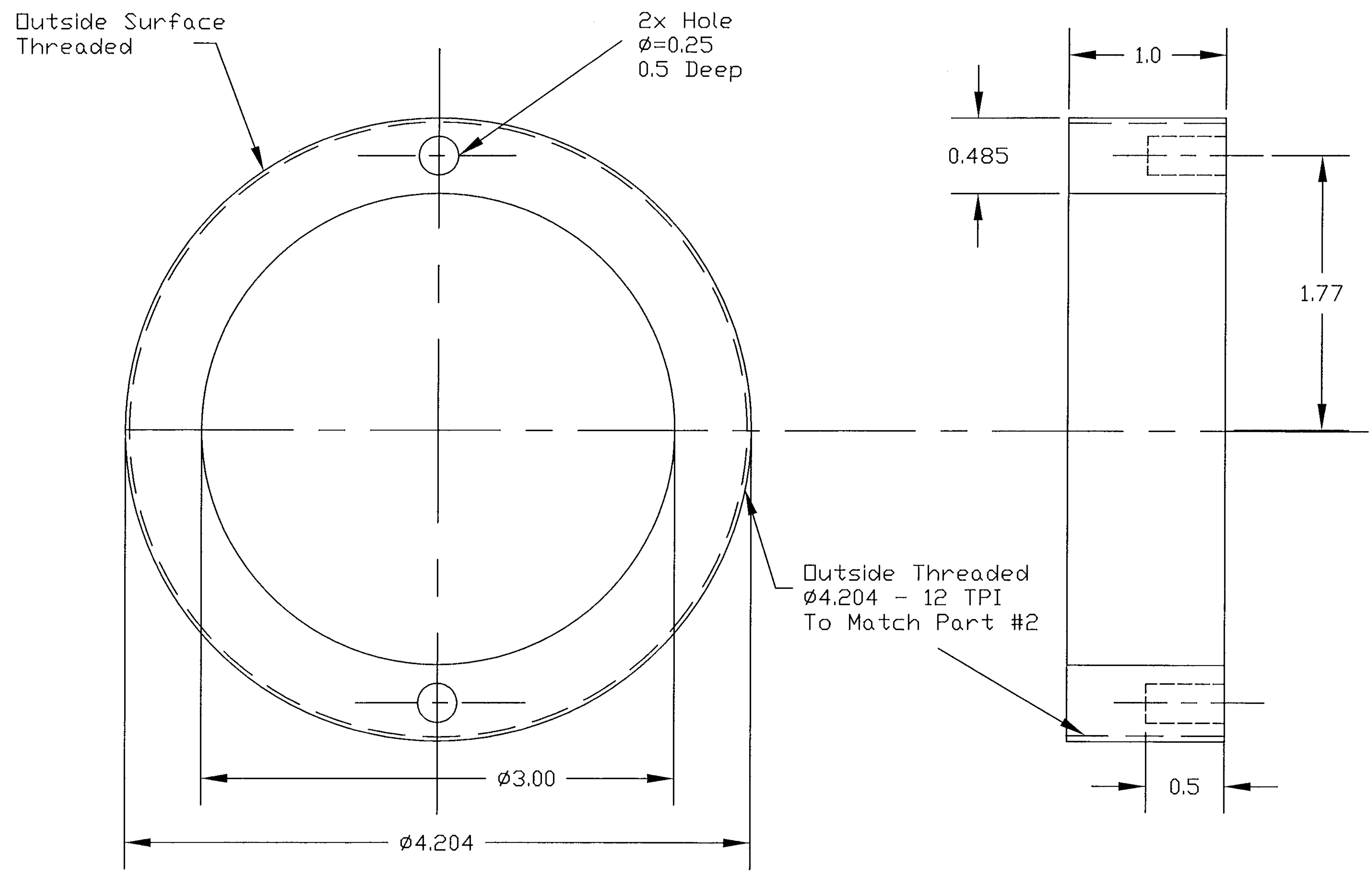


Components

1 Diaphragm Fastener
2 Hull
3 Diaphragm Mount
4 Actuator Mounting Plate
5 Mounting Plate Supports
6 End Cap
7 Electronic Tray Rails
8 Electronic Tray
9 Piston
10 Teflon Ring
11 Diaphragm
12 Linear Actuator
13 O-Ring

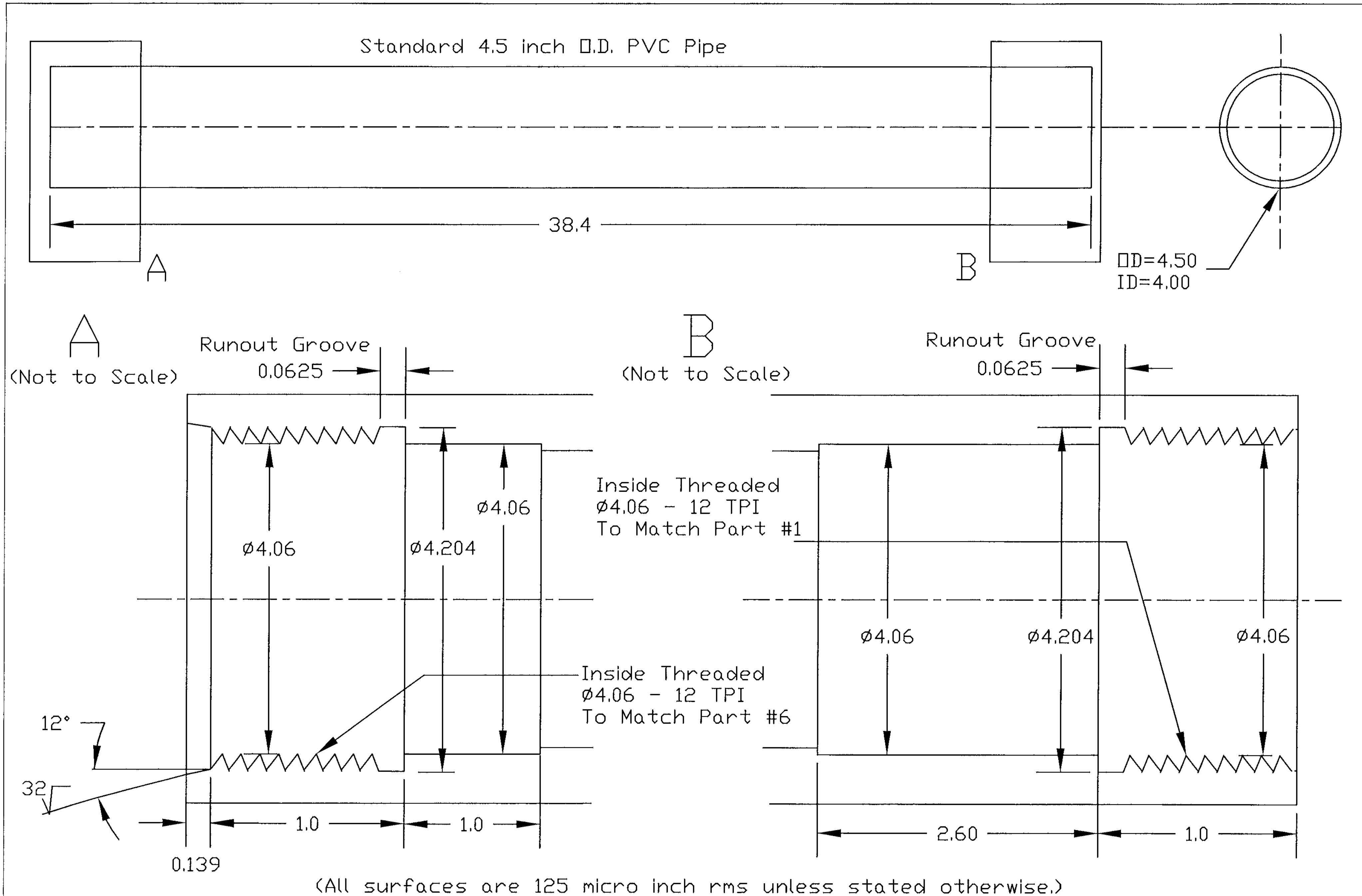
Drawn By: Nick Janes

Date: April 26 2004



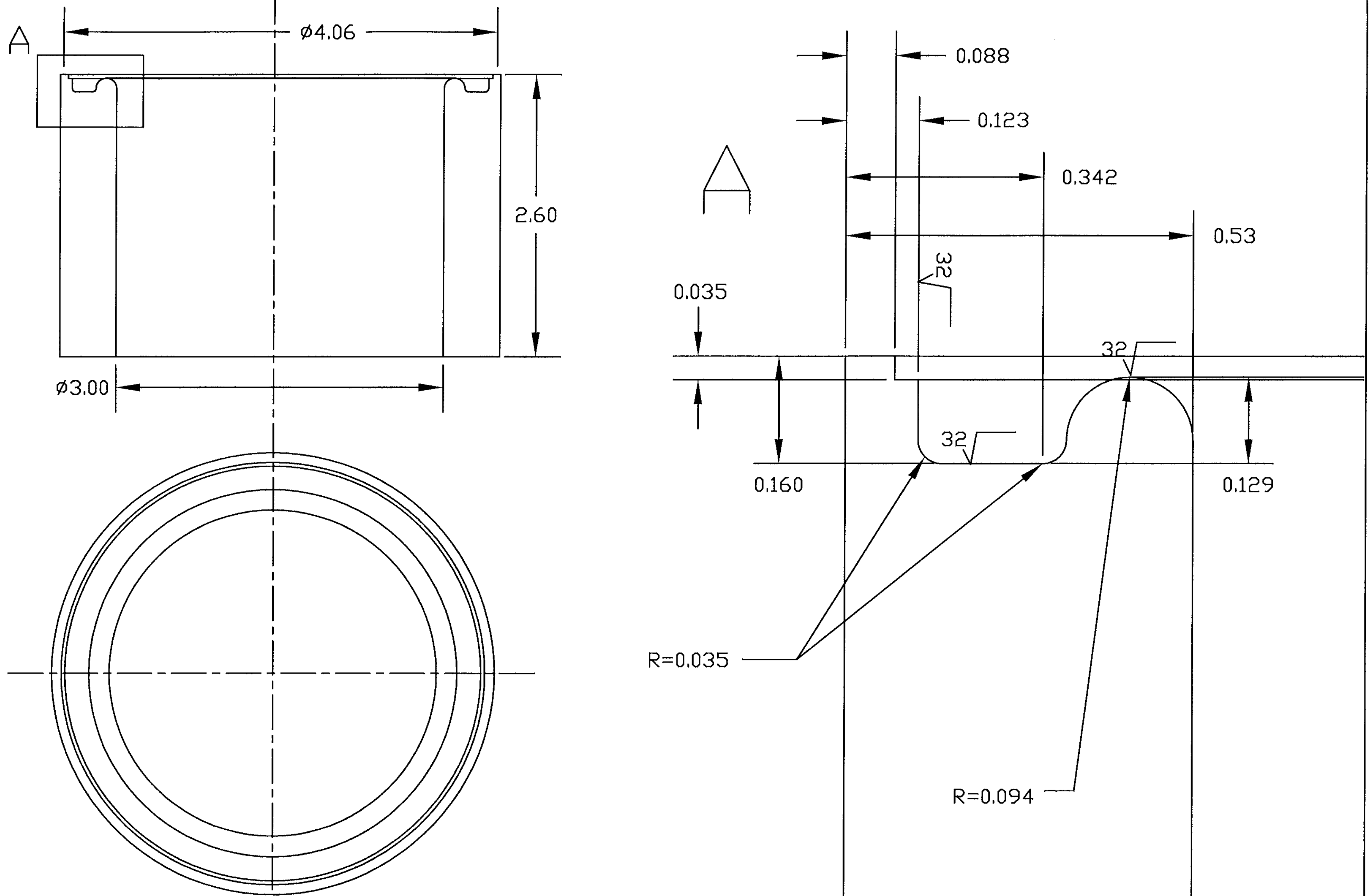
(All surfaces are 125 micro inch rms unless stated otherwise.)

Drawn By: Nick Janes	Part: #1 Diaphragm Fastener	Quantity: 1
Date: April 22 2004	Material: PVC	Dimensions in inches

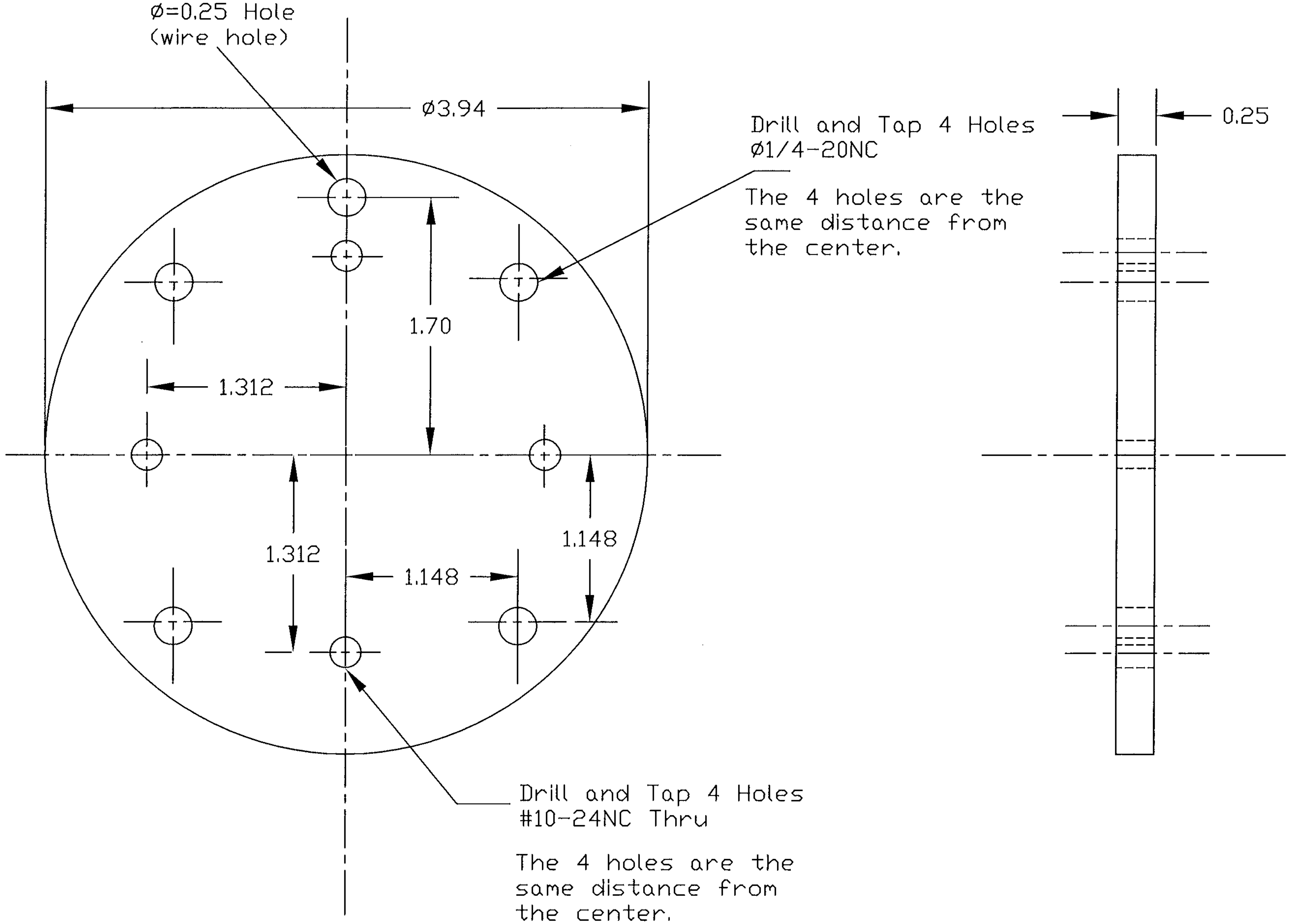


Drawn By: Nick Janes	Part: #2 Engine Hull	Quantity: 1
Date: April 23 2004	Material: PVC	Dimensions in inches

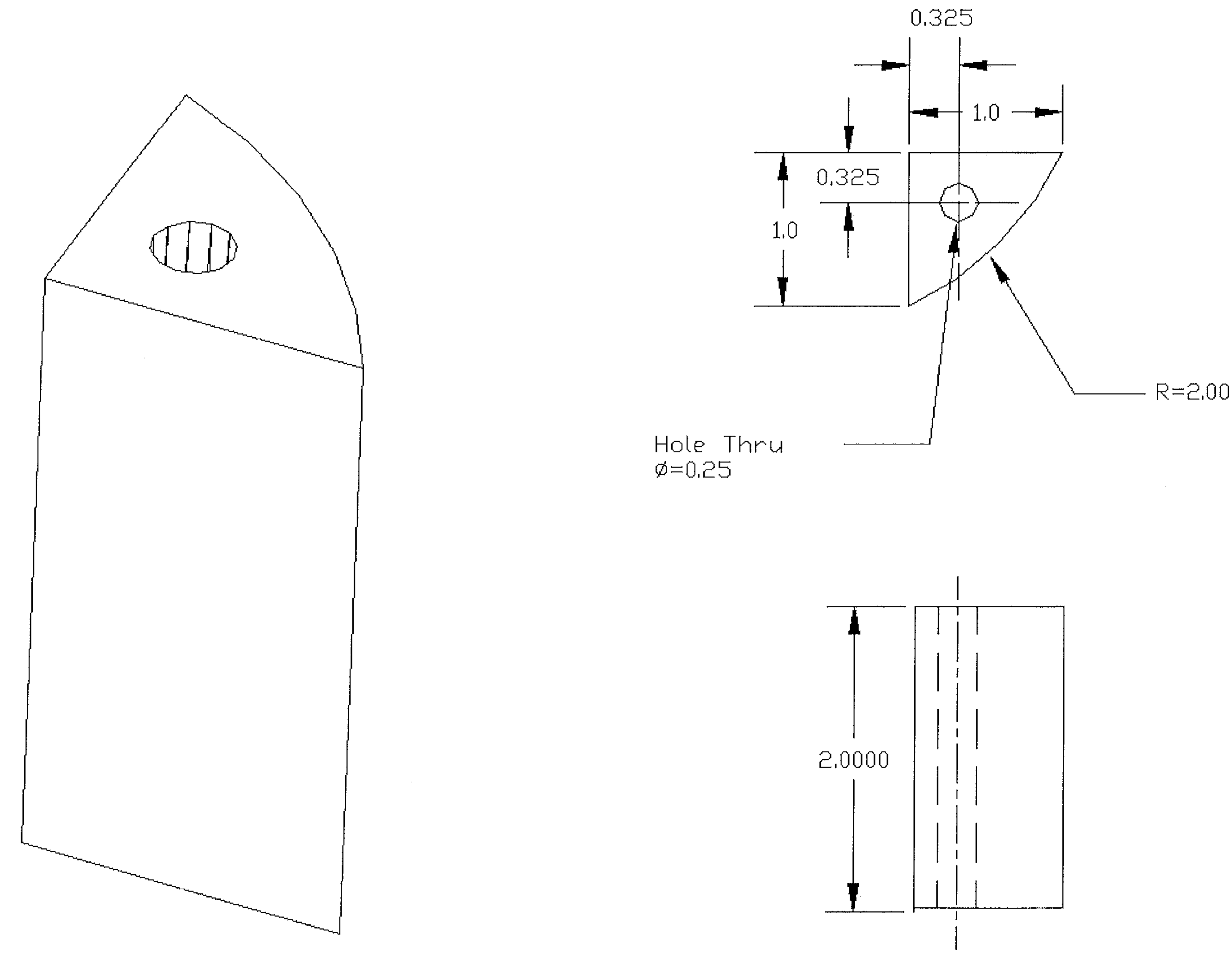
(All surfaces are 125 micro inch rms unless stated otherwise.)



Drawn By: Nick Janes	Part: #3 Diaphragm Mount	Quantity: 1
Date: April 22 2004	Material: PVC	Dimensions in inches

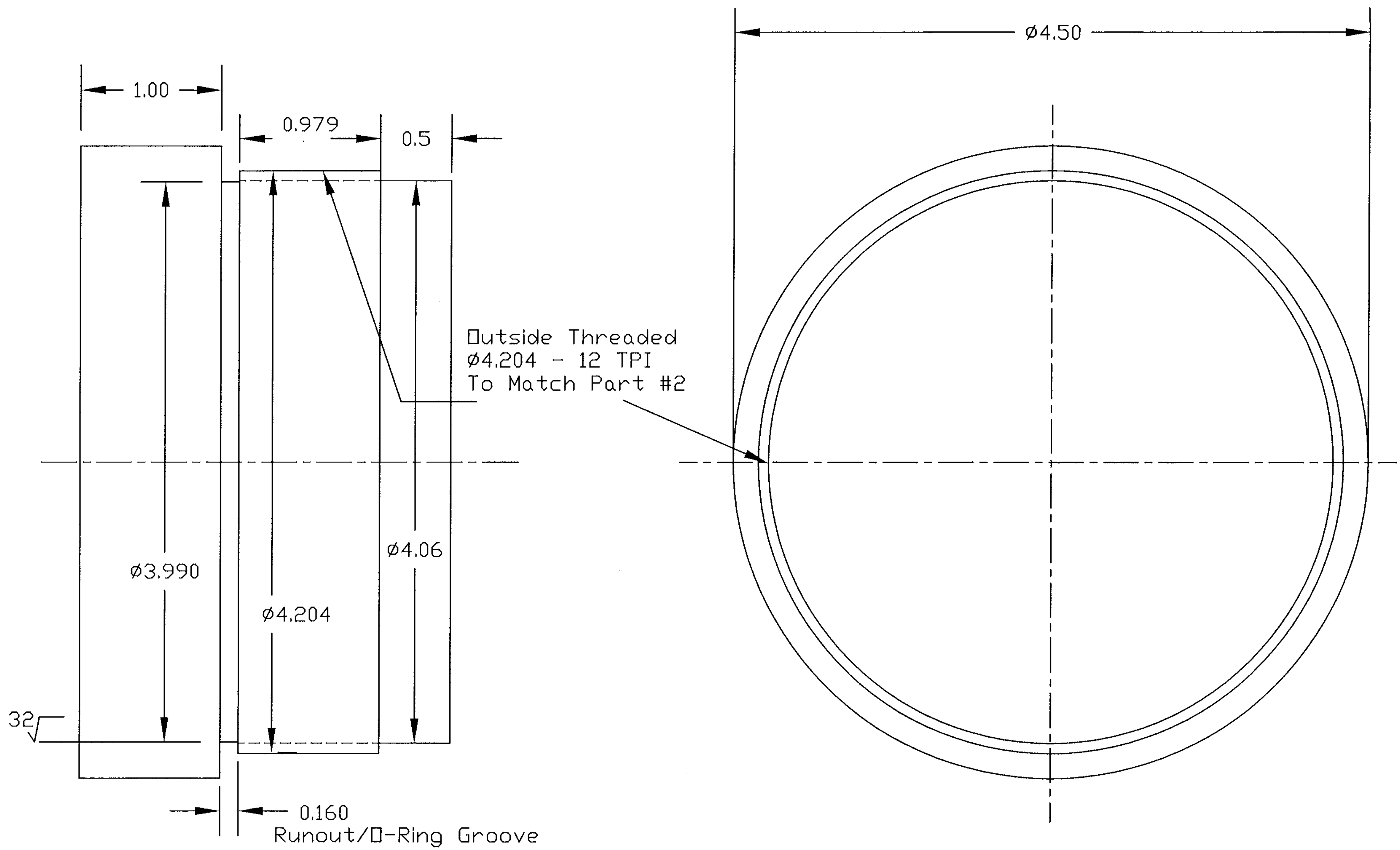


Drawn By: Nick Janes	Part: #4 Actuator Mounting Plate	Quantity: 1
Date: April 22 2004	Material: Aluminum	Dimensions in inches



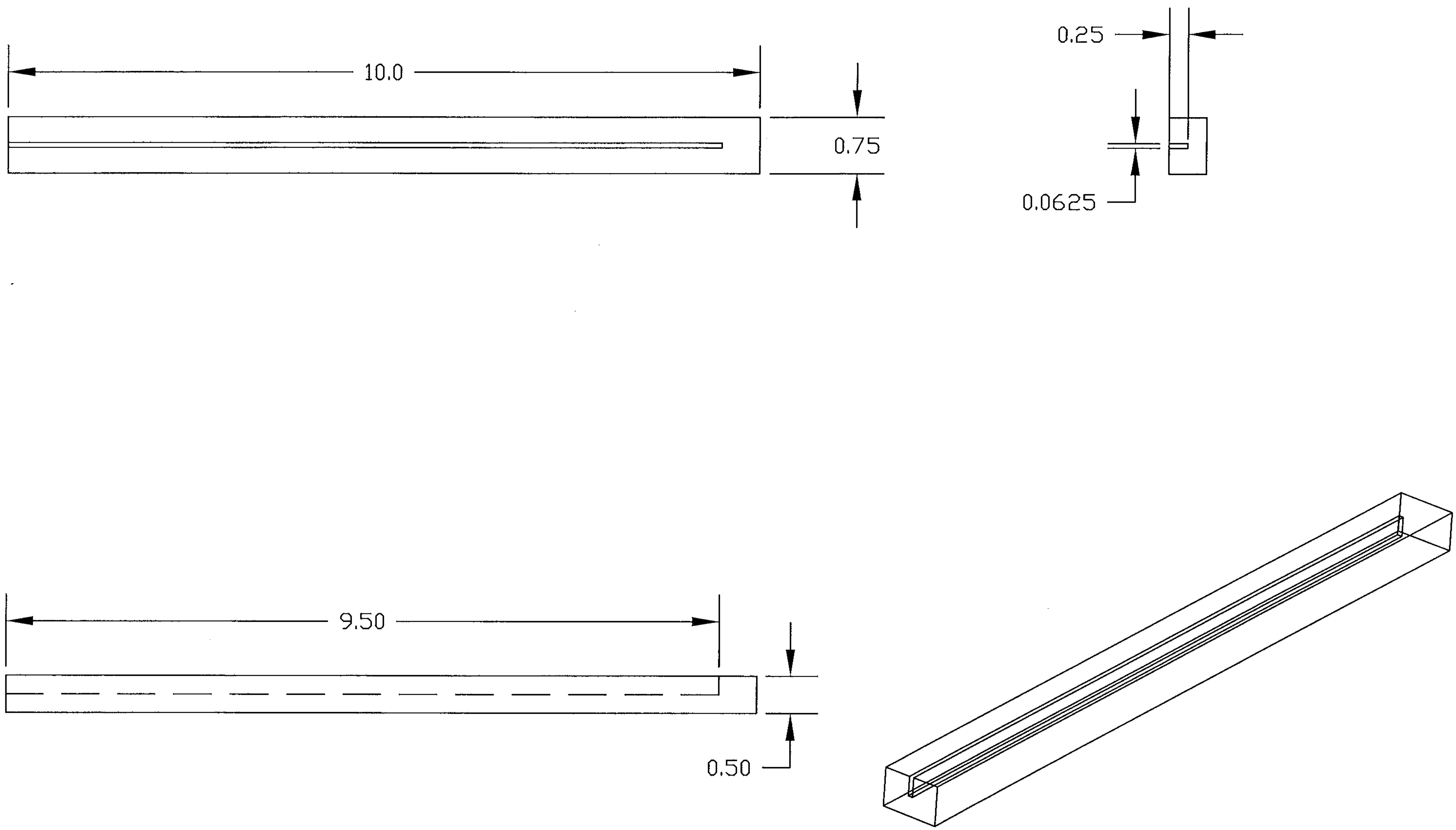
(All surfaces 125 micro inch rms unless stated otherwise.)

Drawn By: Nick Janes	Part: # 5 Actuator Mounting Plate Supports	Quantity: 4
Date: April 22 2004	Material: PVC	Dimensions in inches



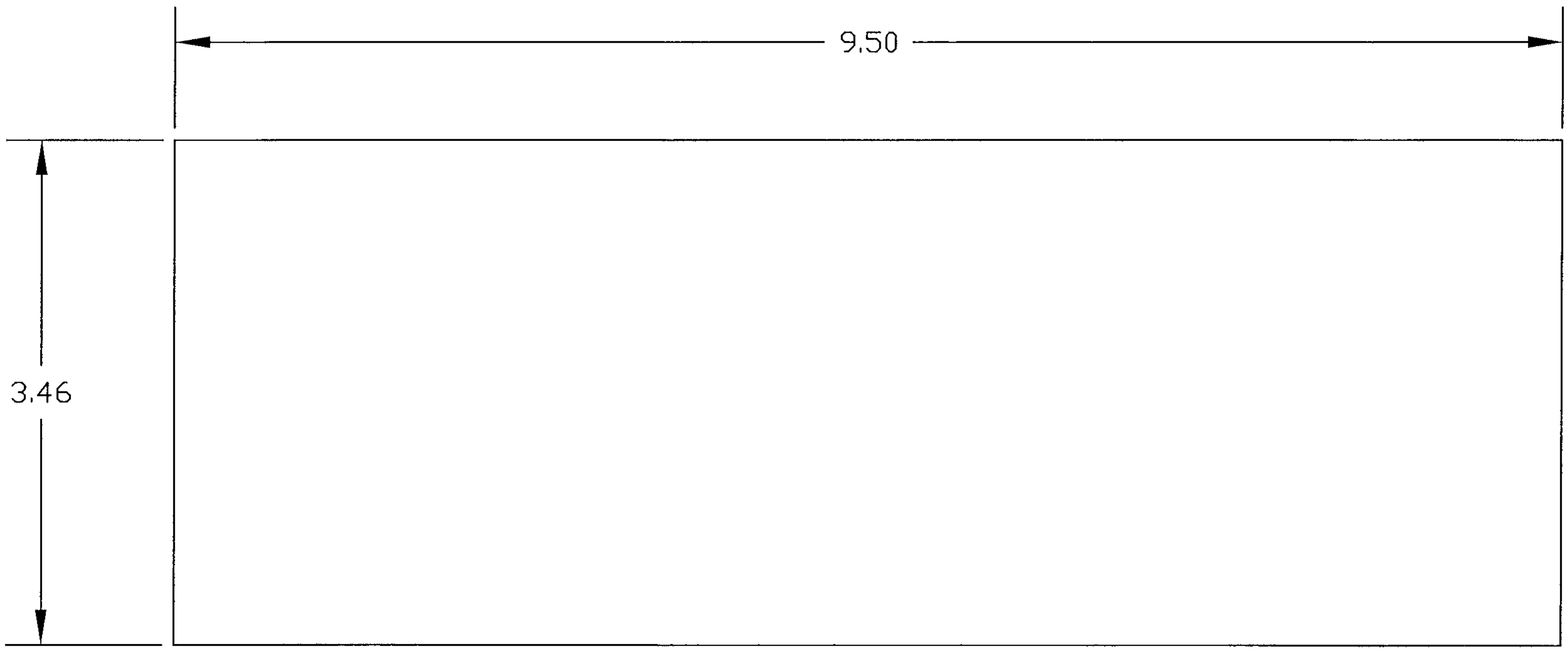
(All surfaces 125 micro inch rms unless stated otherwise.)

Drawn By: Nick Janes	Part: #6 End Cap	Quantity: 1
Date: April 23 2004	Material: PVC	Dimensions in inches



(All surfaces 125 micro inch rms unless stated otherwise.)

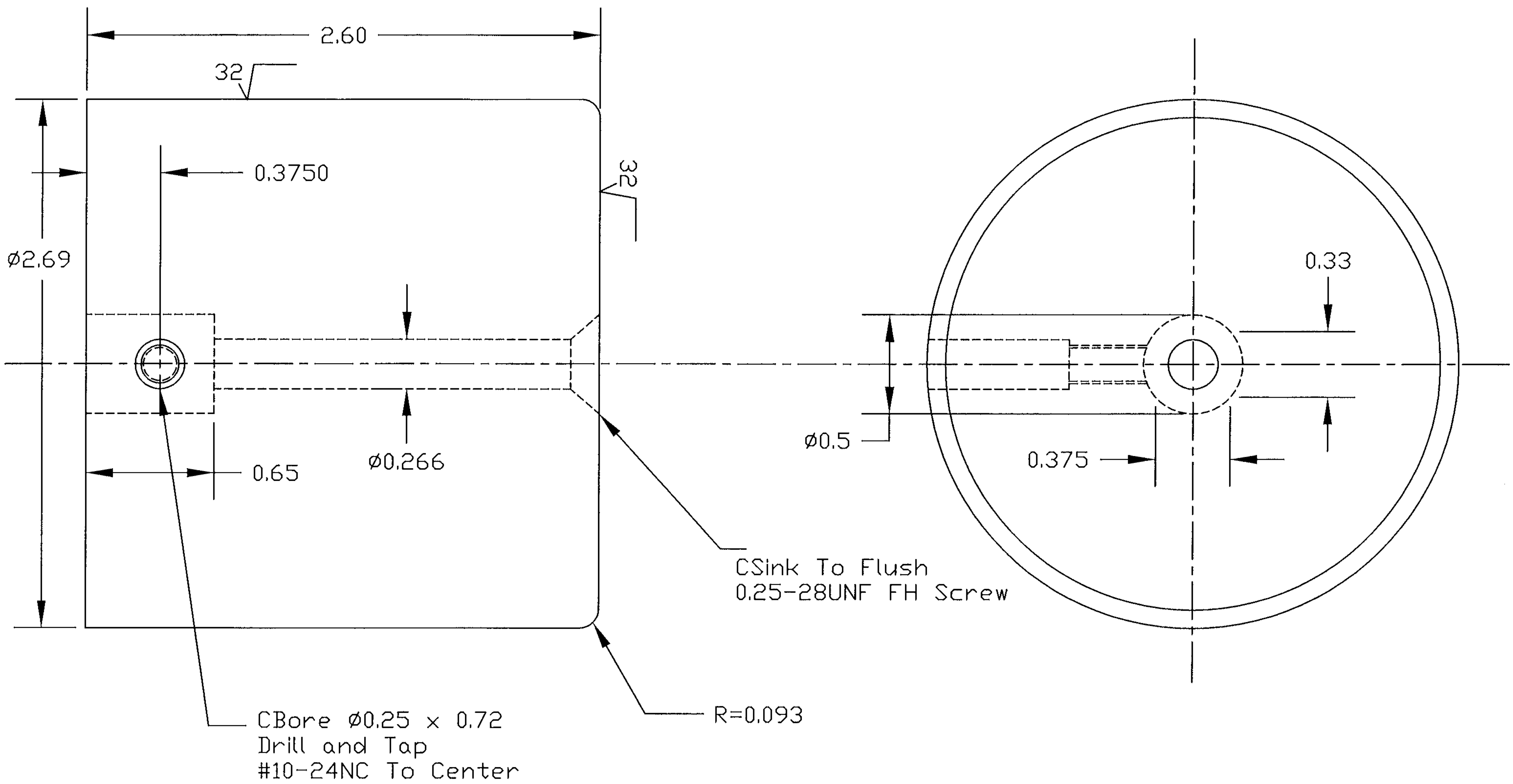
Drawn By: Nick Janes	Part: Part #7 Electronic Tray Support Rails	Quantity: 2
Date: April 26 2004	Material: PVC	Dimensions in inches



1/16 Thick Aluminum Sheet

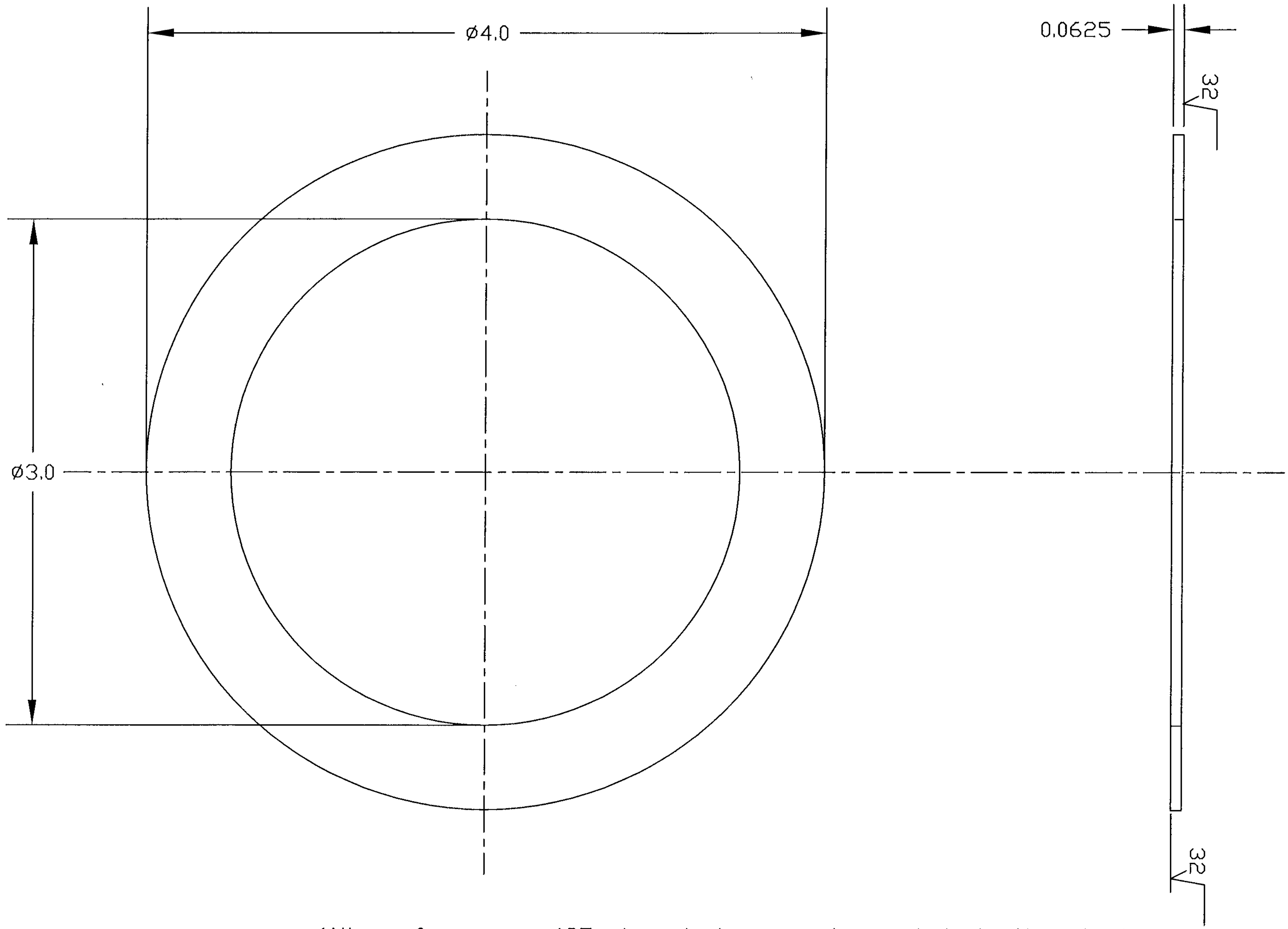
Note: Holes will be drilled into the Electronic Mounting Tray accordingly once electronic boards have been sourced and the mounting patterns have been examined.

Drawn By: Nick Janes	Part: #8 Electronic Mounting Tray	Quantity: 1
Date: April 26 2004	Material: 1/16 Thick Aluminum Sheet	Dimensions in inches



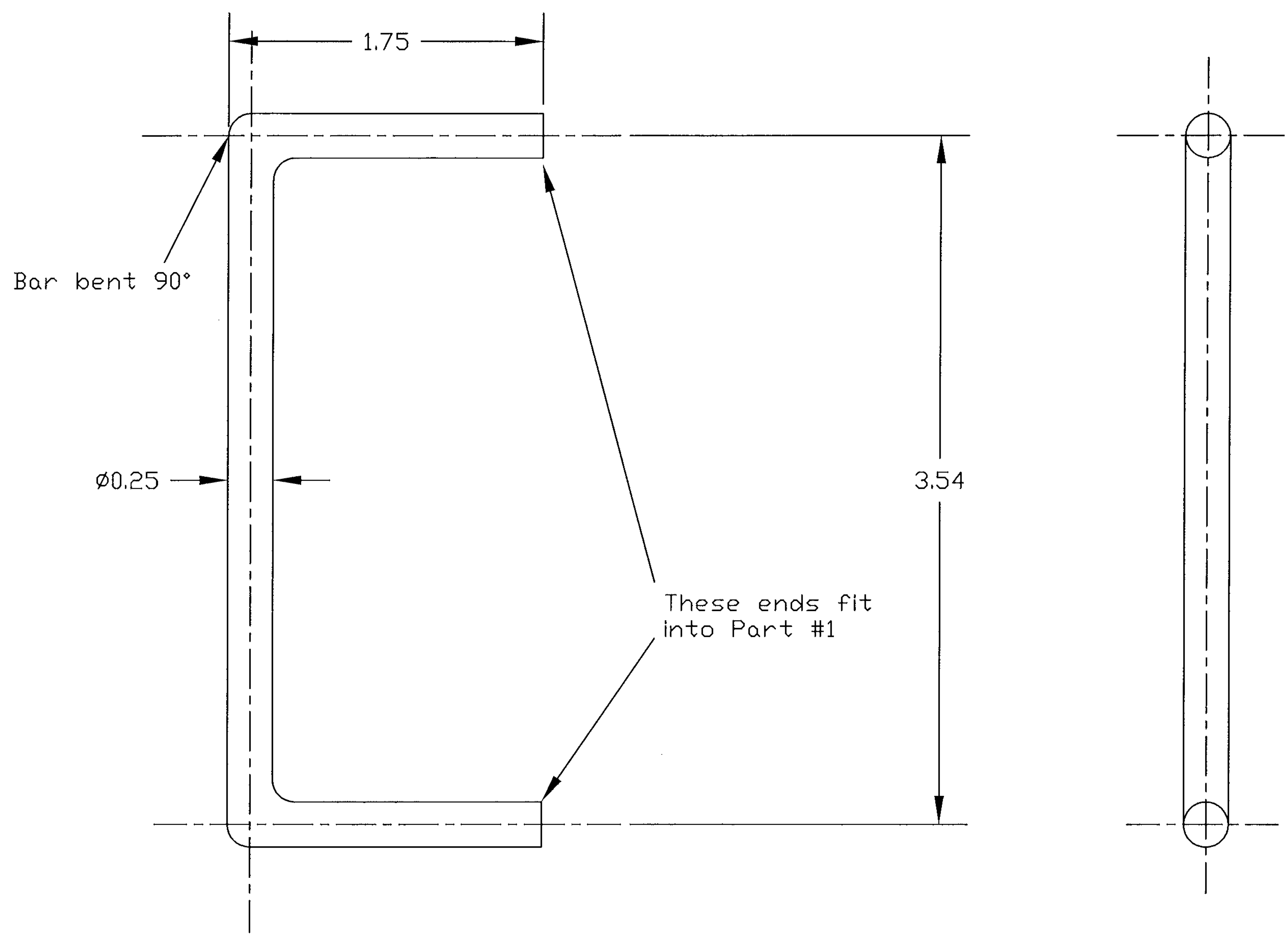
(All surfaces 125 micro inch rms unless stated otherwise.)

Drawn By: Nick Janes	Part: #9 Piston	Quantity: 1
Date: April 26 2004	Material: 304 Stainless Steel	Dimensions in inches



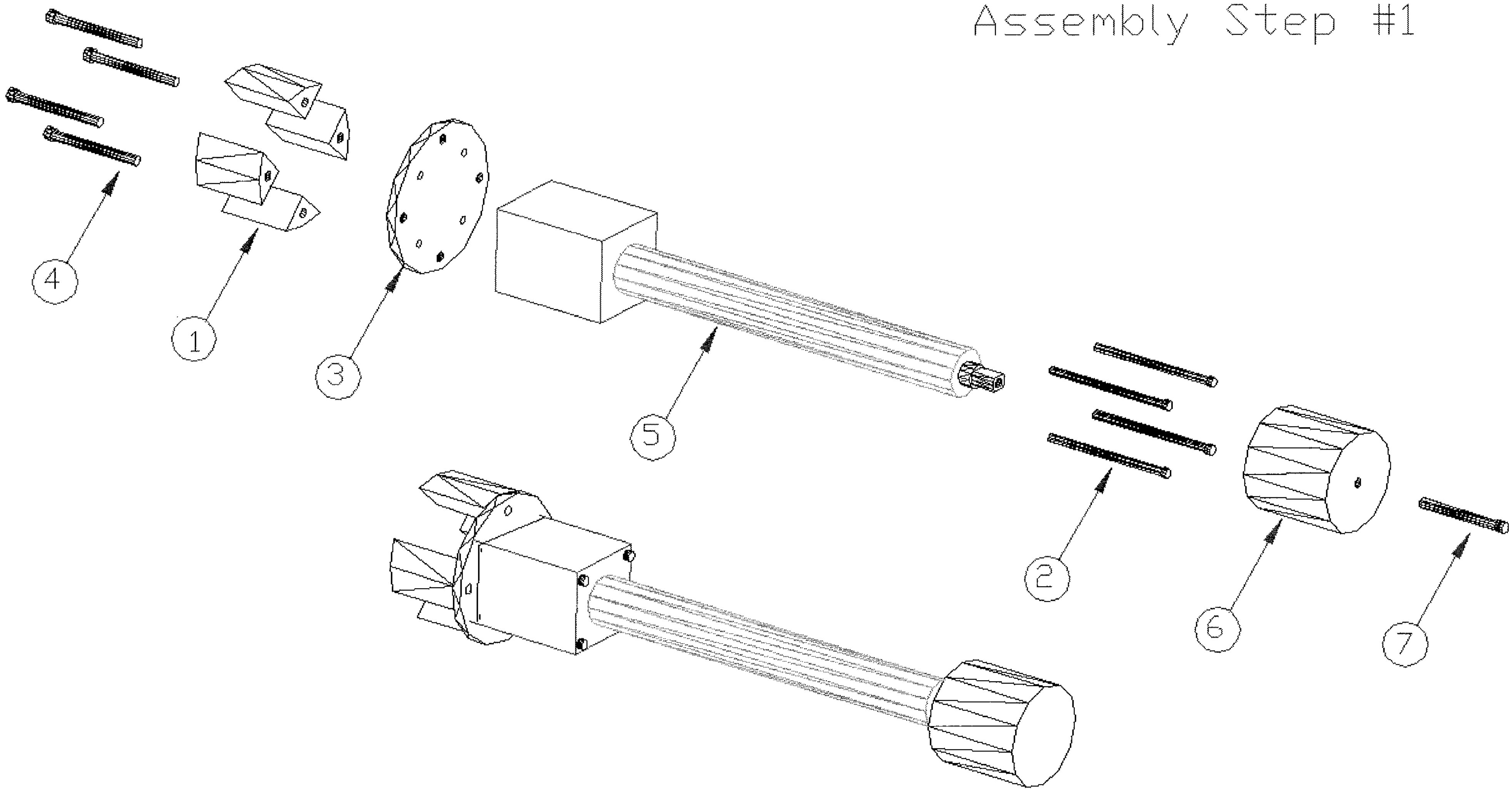
(All surfaces are 125 micro inch rms unless stated otherwise.)

Drawn By: Nick Janes	Part: #10 Teflon Ring	Quantity: 1
Date: April 26 2004	Material: Teflon	Dimensions in inches



Drawn By: Nick Janes	Part: #11 Hand Tool	Quantity: 1
Date: April 28 2004	Material: Ø1/4 Aluminum Bar	Dimensions in inches

Assembly Step #1



Parts

1. Mounting Plate Supports
2. 4x #10-24NC x3 1/2 SH Cap Screw
3. Mounting Plate
4. 4x Ø1/4-20NC x2 1/4 SH Cap Screw
5. Linear Actuator
6. Piston
7. Ø0.25-28UNFx2 3/8 FH
   Hex Socket Screw

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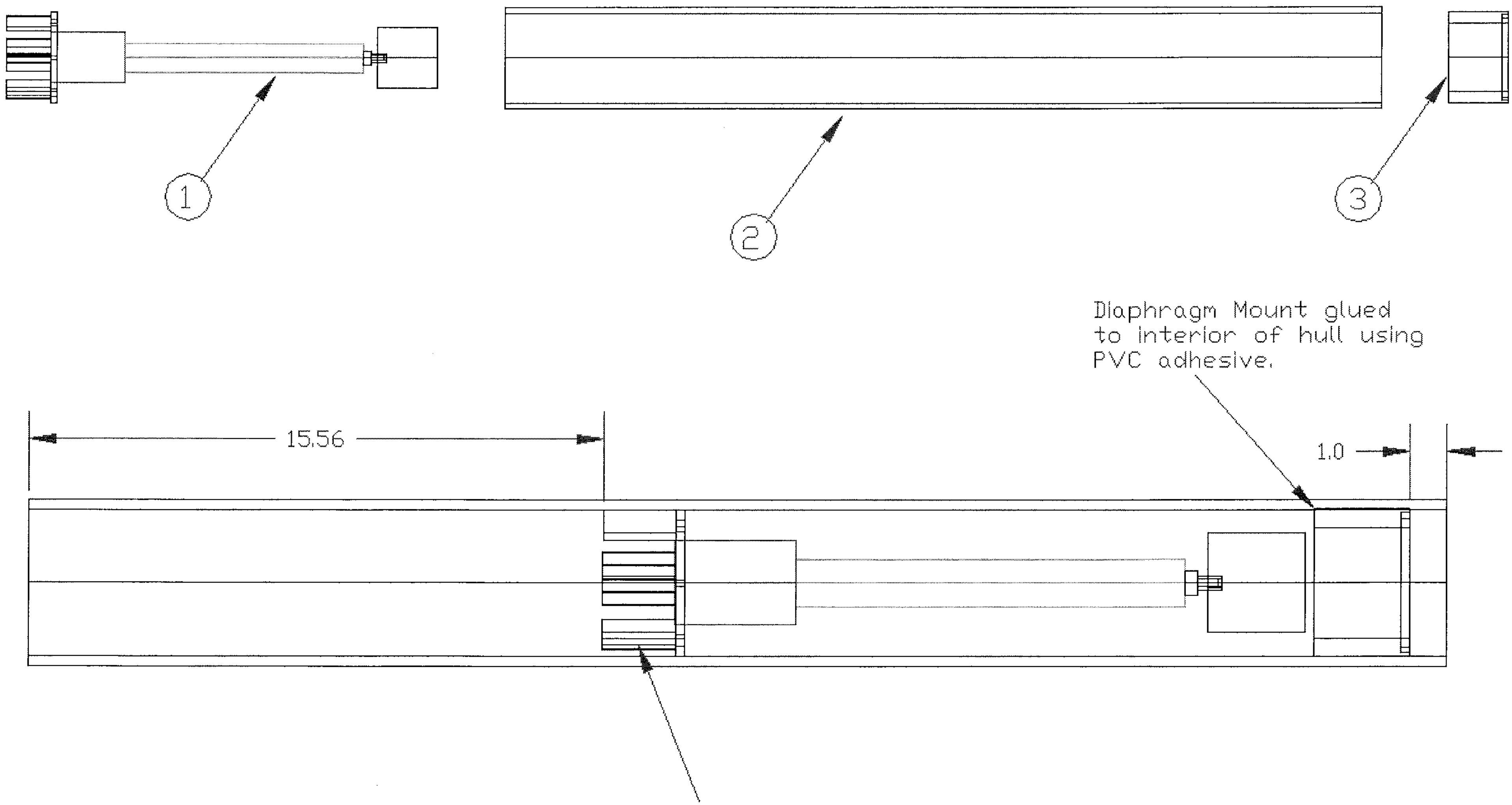
Date: April 28 2004

Assembly Step #2



Parts

1. Assembly from Step#1
2. Hull
3. Diaphragm Mount

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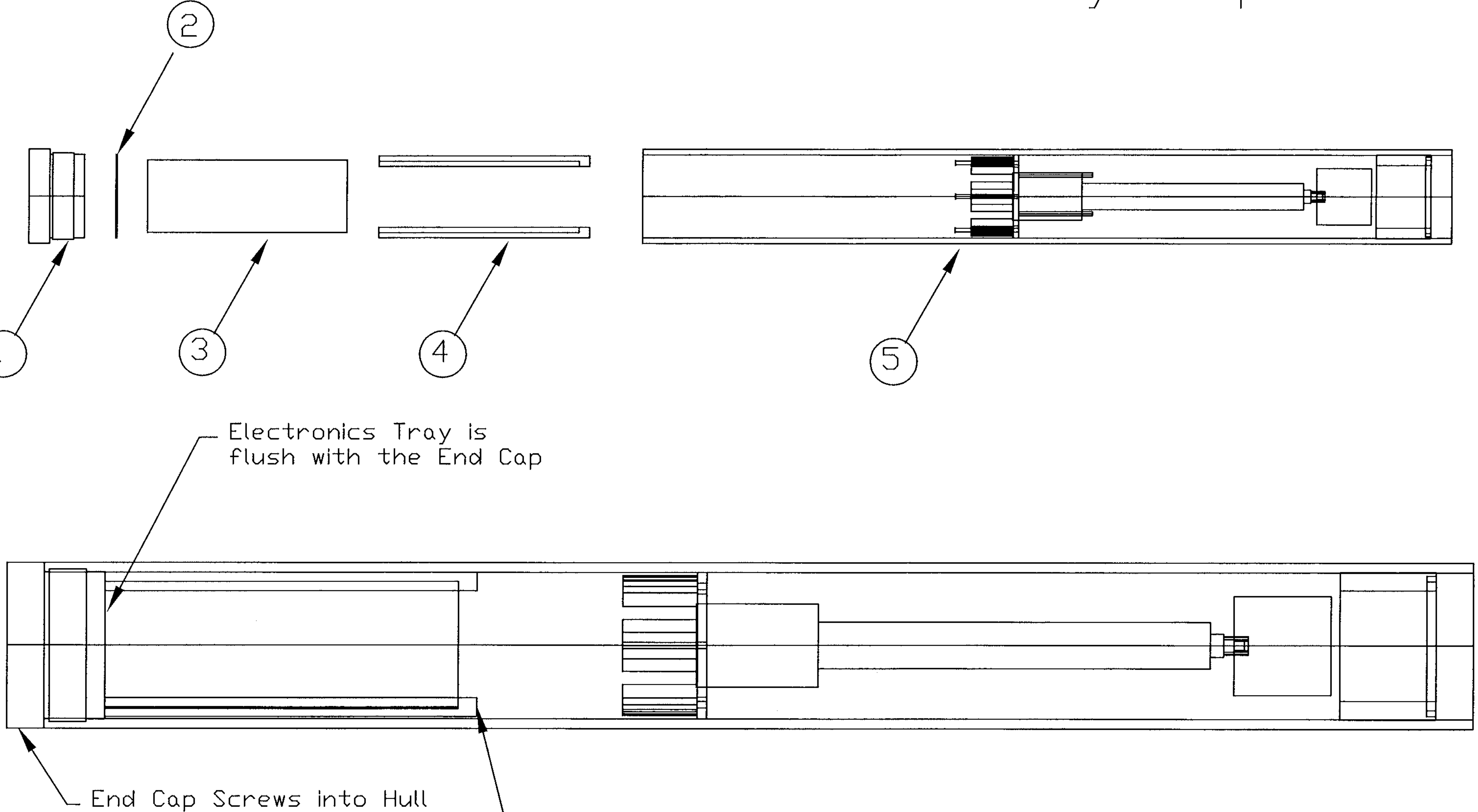
Date: April 28 2004

Assembly Step #3



Parts

1. End Cap
2. O-Ring (-242) Viton
3. Electronics Tray
4. Tray Rails
5. Assembly from Step #2

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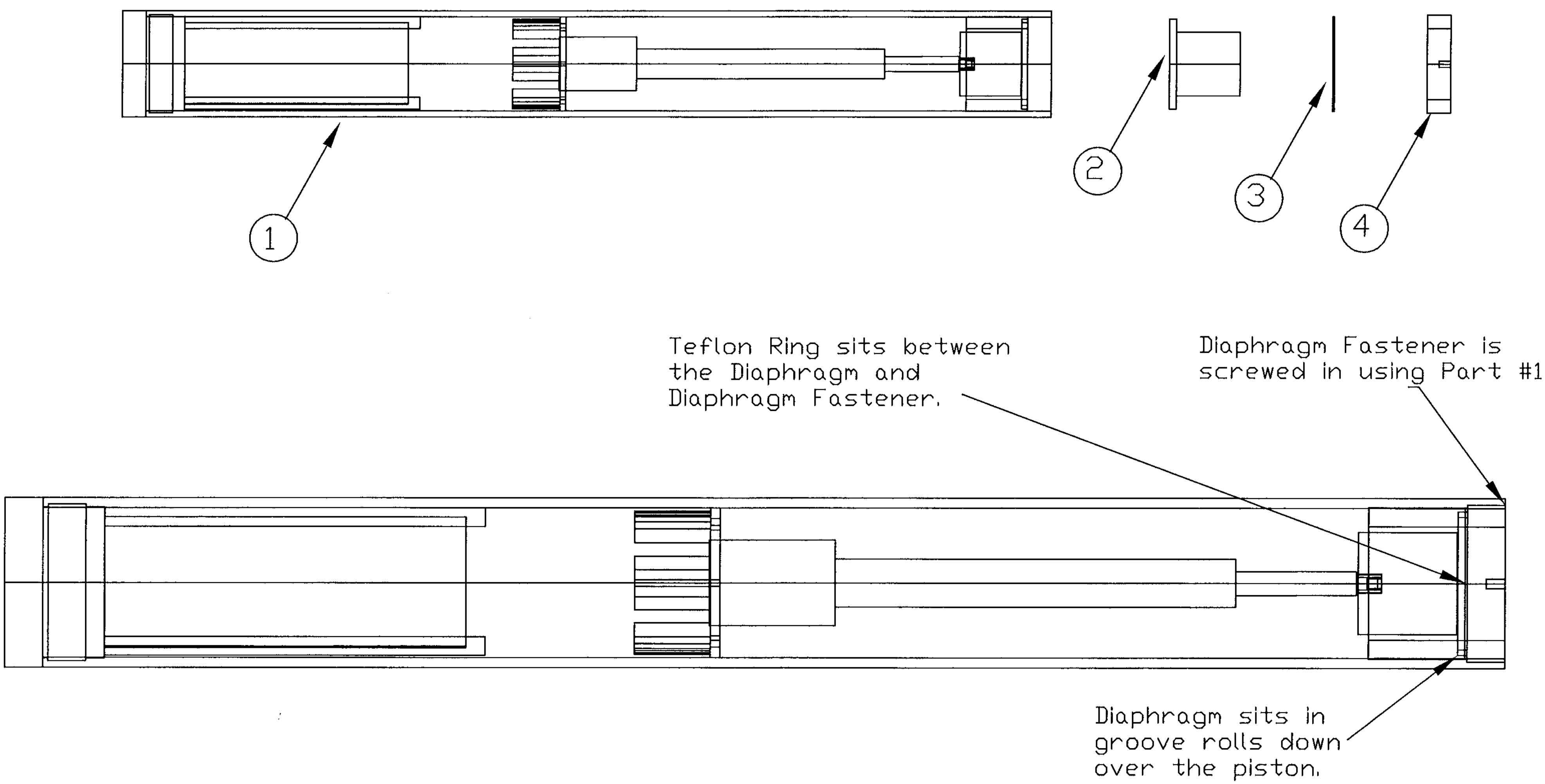
Date: April 28 2004

Assembly Step #4



Parts

1. Assembly from Step #3
2. D-300-300 Diaphragm
3. Teflon Ring
4. Diaphragm Fastener

Drawn By: Nick Janes

Date: April 28 2004

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