A DESIGN OF AN AUTOMATIC BUOYANCY COMPENSATOR FOR A HUMAN POWERED SUBMARINE

by

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Abstract

This thesis will propose a design of an automatic buoyancy compensator for use in a human powered submarine. As the SCUBA tanks in the submarine become lighter, this device will lose buoyancy. This will be accomplished by displacing a volume of water equal in weight to the air in the tanks. Then, as the air in the tanks is used and exhaled out of the craft, the volume of displaced water is decreased, effectively causing the device to gain weight. In this manner the weight of the air used will be compensated for exactly, and the craft will remain at the same weight regardless of the amount of air remaining in the tanks.

Thesis Supervisor: Professor A. Douglas Carmichael
Title: Professor of Ocean Engineering

Dedication

To my mother, of whom I am immensely proud, and my father, who supported and encouraged me along the way.

With thanks to Professor Carmichael, who forgave my lapses in thesis work due to lacrosse, and to Coach Allessi, who forgave my lapses in lacrosse due to thesis work.

Table of Contents

List of Figures

Chapter 1

Introduction

In recent years vehicles powered by alternate sources of energy have gained the attention of the popular press and the public. The solar car race in Australia last year drew entrants from around the world, and all eyes watched as the human powered airplane named Dadelus attempted to recreate the legendary flight across the Agean Sea.

This summer a new human powered endeavor will be undertaken; on June 23rd entrants from around the country will compete in the first ever human powered submarine race. The race will be a head to head competition around a closed course, and the three day tournament will determine a winner through an elimination type format.

There are a number of rules which govern the race and the entrants in it. For example, each submarine must have two people inside: one pilot who steers the craft, and one peddler, who powers it. This guarantees that each craft will have one crew member who is not near exhaustion.

The regulation with which this thesis is concerned specifies a buoyancy limit on the craft. Again for safety reasons, it is desired that the craft always be positively buoyant so that should there be any problems, the craft will bring itself to the surface. Race rules regulate that each craft have two pounds of positive buoyancy throughout the race.

Because of this buoyancy a downward lift must be generated by the control surfaces in order to keep the craft at an even depth. Just as in airplanes, this lift causes drag, causing the submarine to expend some of its precious energy fighting this dissapative effect. Since the drag force will become larger the more buoyant the ship is, it is desirable to keep the buoyancy as close to the two pound limit as possible.

What makes this difficult is that the weight of the submarine will change throughout the race. Both of the crew members will be breathing using SCUBA gear. The oxygen supply tanks are typically filled with air at 3300 pounds per square inch. As the air is used the tanks get lighter, and the weight change is significant in the context of this race. For example, a 71.4 cubic foot tank rated for 3000 psi, which is usually filled to 3000+10% or 3300 psi, has a buoyancy of -9.5 pounds full and -2.3 pounds empty, a change of over six pounds. Thus, near the end of the air supply the craft will be fighting as much as four times the lift induced drag that it did at the start of the race.

This thesis will propose a design of an automatic buoyancy compensator which will lose buoyancy as the tanks become lighter. This will be accomplished by displacing a volume of water equal in weight to the air in the tanks. Then, as the air in the tanks is used and exhaled out of the craft, the volume of displaced water is decreased, effectively causing the device to gain weight. In this manner the weight of the air used will be compensated for exactly, and the craft will remain at the same weight throughout the race, the minimum two pounds positively buoyant.

Chapter 2

General Design

Structurally, the device consists of two fixed plates at either end with a floating plate in the middle (see figure 2-1). The two end plates are fixed by six rods running between then, which also serve to guide the floating plate as it travels back and forth between the plates.

Between the front plate and the floating plate are a pair of bellows which displace water. Running from the floating plate to the back face of the device are a set of springs which fit over the connecting rods. Acting against these springs is a piston filled with air from the SCUBA tanks. The relationship between the springs and the piston determines the position of the middle floating plate, thus controlling the amount of water displaced by the bellows.

2.1 Determination of Tank Weight

A high pressure line from the SCUBA tanks will be available, which will allow for the monitoring of the pressure inside the tanks. From this information the weight of the tanks will be determined.

The rate of air leakage from the tanks is fairly low, in spite of the high level of activity of the crew. A 71.4 cubic foot tank usually lasts a diver from 45 to 60 minutes. Because of the pedaler's extremely high level of physical exertion the air is expected to last a significantly shorter time, but still at a thermodynamically slow rate.

In addition, since the tank will be immersed in water, the rate of heat transfer into and out of the tank will be high.

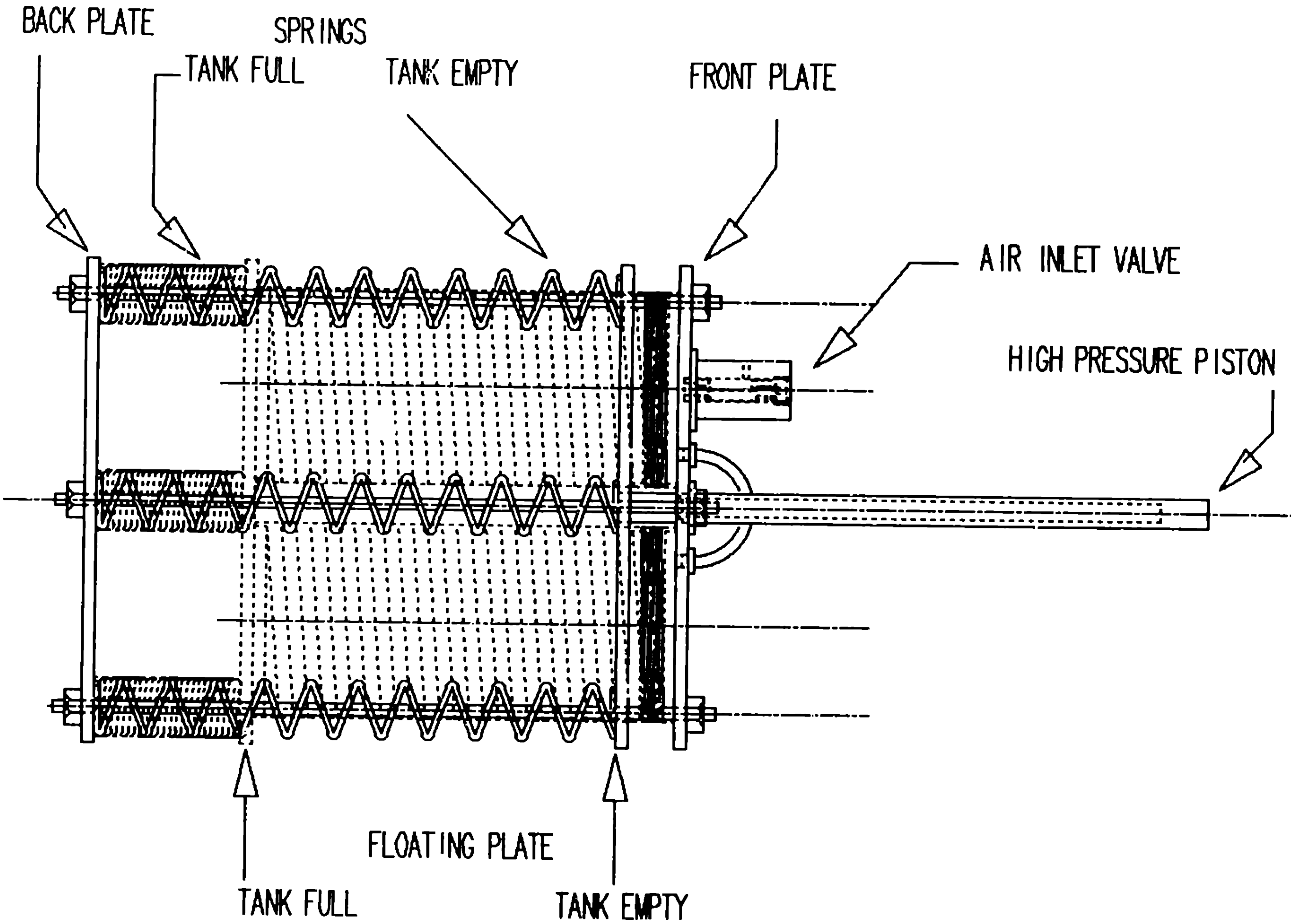

Figure 2-1: Major components of the buoyancy compensator

Due to both the low rate of gas leakage and the high rate of heat transfer in the water, the system can be modeled as isothermal, greatly simplifying the problem.

Under these assumptions, the gas in the tank can be modeled using the Ideal Gas Law, which states:

$$PV=nRT$$

where P is the pressure of the gas, V is the volume of the container, n is the number of moles of gas contained in the system, R is a constant, and T is the temperature. With V, R, and T constant, the leakage of gas from the tanks, which is seen as a change in n, is directly proportional to the pressure of the gas. Thus, a direct linear relationship between the pressure and the weight of the gas remaining in the tanks can be expected.

Using a linear spring and a piston to measure the pressure of the gas remaining in the tanks allows for the transformation of this pressure into a displacement. The direct relationship between pressure and weight and the linear characteristics of Hooke's Law mean that the weight of the gas is linearly related to the position of this piston.

Because of the high friction O-rings would cause were they used to seal the piston, it was decided to avoid their use. Instead, the clearance between the piston and the surrounding walls is extremely tight, limiting the air leakage around the piston. Also, since the piston is a solid rod, the length of this tight clearance is quite long, also contributing to low leakage (see figure 2).

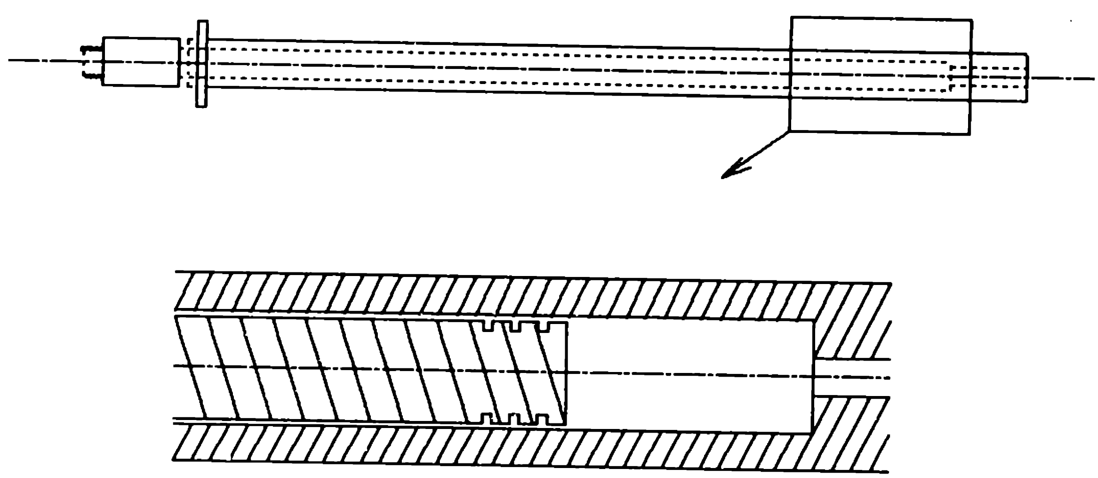

Figure 2-2: Piston, and highlight showing notches

In addition to the tight clearance, a number of notches around the piston wall will be used to limit the leakage of air (see exploded view of figure 2-2). Because of the tight clearances and the high pressure difference from one end to the other, flow through this opening will be in the supersonic relm. The addition of notches along the piston increases the number of accelerations and deceleration of the leaking air (see figure 2-3), dissipating energy in the swirls which form in each notch, resulting in an acceptable flow flow of air between the piston and the cylinder wall.

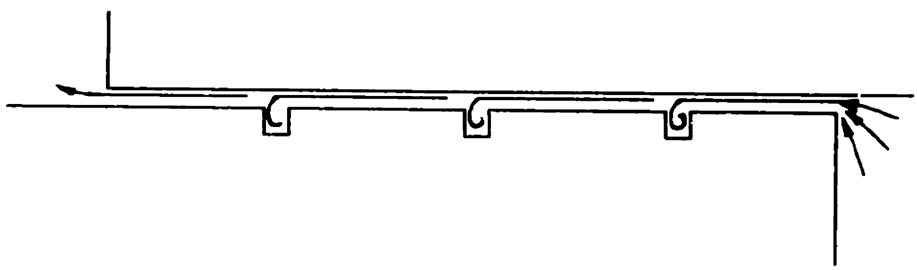

Figure 2-3: Supersonic flow through notches

The section of the piston rod which never enters the cylinder is wider than the rest of the piston rod. This can be seen in figure 2-2. This is necessary because the rod is loaded near the critical buckling load. This wider section reduces the length of the rod with a diameter of .396 inches, increasing the rod's resistance to buckling.

2.2 Water Displacement

The approach to water displacement was one of the most debated areas in the conception of this design. The only requirements was that the volume be completely controllable. Concepts which were considered included pistons, bladders, fabric bellows, metal bellows, and rotary pistons.

2.2.1 Displacement Volume Size

The SCUBA tanks for which this device has been designed are standard tanks made by ScubaPro. The tank is made of heat treated steel and holds 71.4 cubic feet of air at standard pressure. The tank is rated by the Department of Transportation at 3000 psi, but it is normal practice to overfill SCUBA tanks by 10%, to a pressure of 3300 psi. The buoyancy of this particular tank is -9.5 pounds when filled to 3300 psi and -2.3 pounds when empty. Thus, the change in buoyancy of this tank is 7.2 pounds.

Since sea water will be taken on as ballast, knowing the appropriate volume is necessary. One cubic foot of sea water weighs 64 pounds, so .1125 cubic feet of sea water is needed to gain the weight the scuba tanks will lose. This is 194.4 cubic inches.

194.4 cubic inches corresponds to a cube just under 6 inches per side. In the shape of a cylinder, one with a diameter of 8 inches will have a length of 3.9 inches, and one 6 inches in diameter will have a length of 7.0 inches. Thus, the volume of water is of a small and manageable size.

2.2.2 Displacement Method

For volumes of the approximate dimensions quoted above none of the designs considered could withstand high pressures within the displacement volume. Certainly the bellows would burst if filled with 3300 psi air, but even a piston-type displacement volume would need prohibitively thick walls to withstand the pressure. Thus, the design arrived at was a low pressure displacement volume whose volume is controlled by the position of a small, high pressure piston. Under these expectations, the bellows designs were considered the best, and the design continued investigating bellows made of both metal and fabric.

One question which fabric bellows raised is their inability to withstand positive pressure differences between outside and inside pressures without collapsing. The submarine is expected to achieve depths of at least fifteen feet, and it may dive as far as thirty feet. The will cause pressures of up to one atmosphere across the bellows, under which a fabric bellows would likely collapse.

For this reason metal bellows were investigated. These are bellows made of thin steel shaped and welded together. Bellows of the necessary size for this project could withstand pressure differences of up to twenty psi, so regulating the pressure difference across the bellows would not be as critical.

However, metal bellows are designed to meet extremely high specifications. Most of their applications are in aviation and the containment of radioactive gas, and their requirements are much more stringent than those stipulated by this design problem. These characteristics make metal bellows prohibitively expensive, ruling them out of the design.

Thus, a fabric, bellows-type displacement volume was chosen. An air hose designed for use venting air from clothes driers was chosen as an off-the-shelf bellows. Shape is held by a continuous wire wrapped in helical fashion, and a vinyl tape is wrapped along the coil between turns of the wire. Due to this construction, the bellows collapses to less than 7 percent of its extended length.

2.2.3 Configuration

The diameter of this hose, however, is only four inches, so a length 15.5 inches long is needed to displace 194.4 cubic inches of water. A stroke of this length was avoided by putting two of these bellows between two parallel plates. Having these two cylinders in parallel reduced the stroke necessary for a change in volume of 194.4 cubic inches to 7.75 inches. The layout of the bellows, the springs, and the piston can be seen in figure 4.

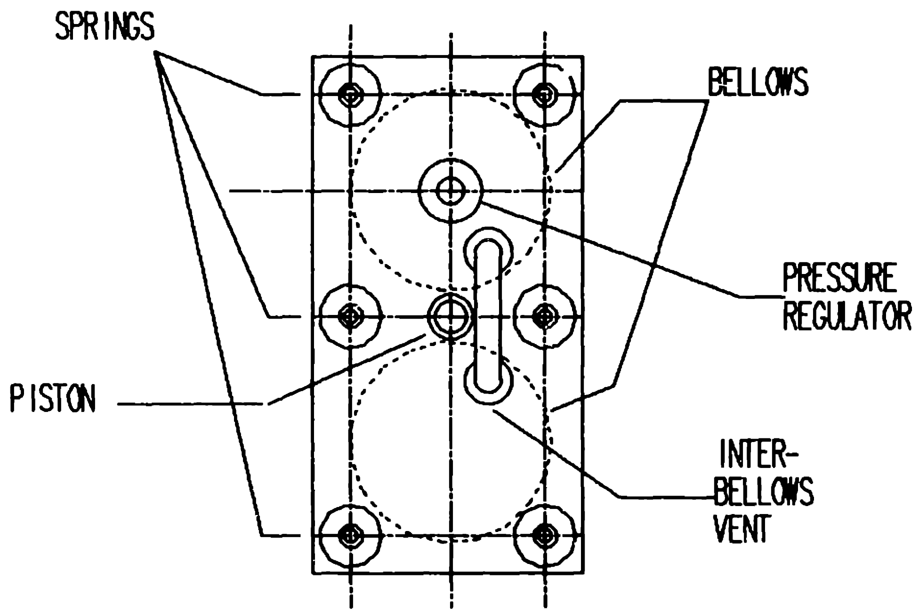

Figure 2-4: Layout of springs, piston, and bellows

2.2.4 Resistance of Pressure Differences

Although the wrapped coil provides some resistance to higher outside pressures, the device may still encounter pressures great enough to crush these bellows. In order to keep the pressure difference across the bellows low enough to prevent their collapse, a regulator was added to the design. This regulator keeps the pressure inside the displacement volume equal to the ambient pressure. The two cylinders were vented with each other, allowing pressure regulation to be accomplished with one set of regulators.

2.3 Pressure Regulation

Pressure regulation inside the displacement volume can be seen as two separate cases. In one case the pressure inside the bellows is higher, as would be the case when the submarine is ascending. On the other hand, when the craft descends, the pressure inside the bellows tends to be lower than the ambient pressure of the water. Each of these cases is handled separately.

2.3.1 Pressure Relief Valve

High pressure inside the bellows threatens to cause them to burst. To prevent this, a pressure relief valve is used to vent the bellows when the pressure difference above the ambient water pressure reaches too high a level.

A hole venting the bellows to the sea is covered on the outside by a simple rubber flap which is fastened at both ends (see figure 5). When the outside pressure is greater than that inside the bellows, the flap is held over the hole by hydrostatic pressure. On ascent, however, the ambient pressure of the water will decrease, and the flap will be forced open by the higher pressure inside the bellows, thereby venting the air into the sea. When the pressure difference has decreased to an acceptable level again the rubber flap will once more cover the hole.

2.3.2 Air Intake Valve

On descent the ambient pressure becomes greater, threatening to crush the bellows arrangement. The time constant of any pressure control system would have to be quick enough to prevent a dangerously high pressure difference from developing during a high speed descent.

One approach which was considered was allowing a small constant leakage into the bellows. When descending this leakage would fill the bellows until the pressure was

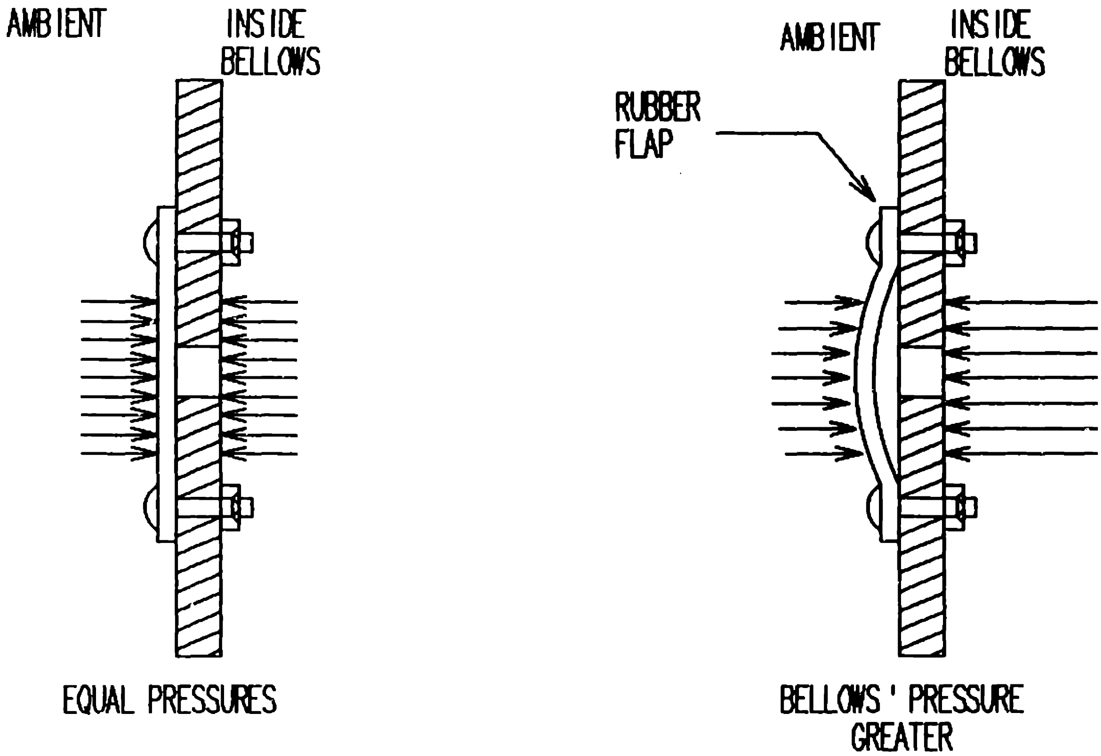

Figure 2-5: Pressure Relief Valve

equalized, at which point excess air would be released through the flap valve described above.

This design would be leaking air all the time; on an even plane, or even ascending, air would leak into the displacement volume and be vented througn the pressure release flap valve. Of course, this air is wasted, giving two competing requirements: one is fast response to a descent, preventing the collapse of the bellows, and the other is not to waste precious air.

Unfortunately, there is no accurate way of predicting the maximum rate of decent of the craft, and thus, nothing but a guess as to the rate at which air might have to be vented into the displacement volume. It is also unclear how much air will be used during the race, so the balance between these two competing requirements is very difficult to find.

Instead, a pressure regulator was chosen. While the standard, two stage, SCUBA regulator was considered, the desire to keep the number of pressure lines to the device at a minimum weighed against this choice. The standard SCUBA regulator would use a low pressure (100 psi) line from the tank's first stage regulator to the final stage regulator, which would be located at the displacement volume. Since this would be in addition to the high pressure line with which the tank pressure is measured, two lines to the device would be necessary, decreasing, by some measure, reliability.

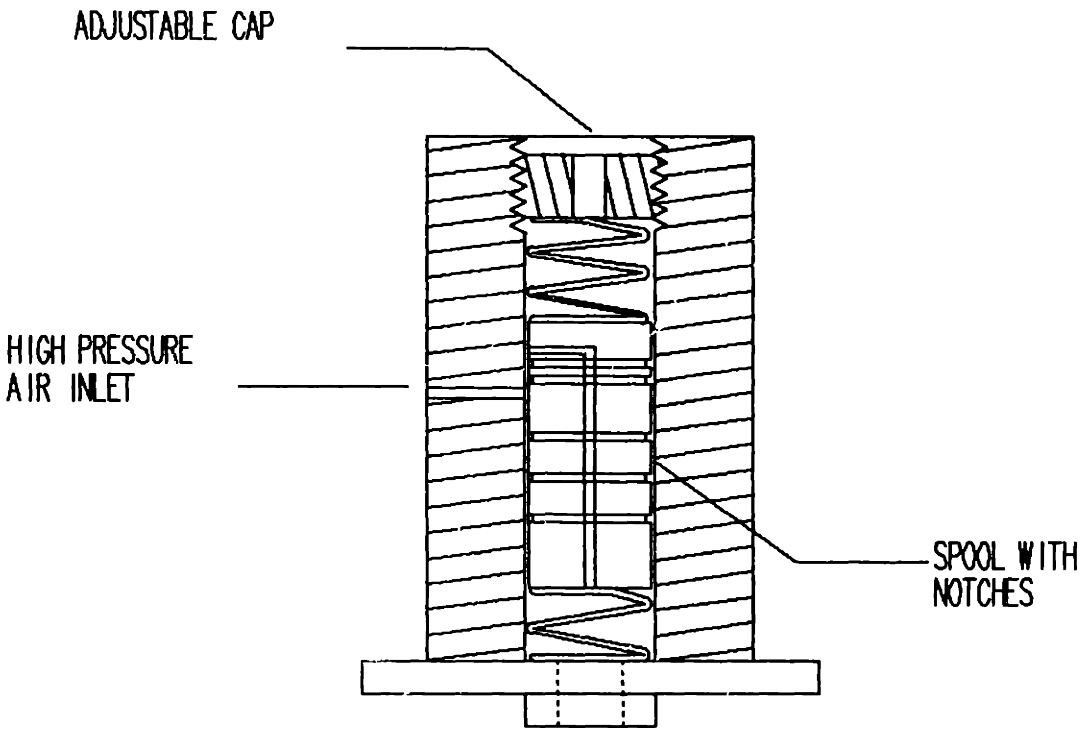

Figure 2-6: Air intake valve

Instead, a small and simple spool type valve was designed (see figure 6). The ambient pressure acts on one end of the spool, and the pressure inside the displacement volume acts on the other end. When the pressures are equal the spool is centered between the two springs. If the ambient pressure is greater than the displacement volume pressure, the piston is forced inward, allowing air to be vented into the displacement volume, equalizing the pressure.

Air flow around the spool as regulated in the same manner as flow around the piston. In this device, which must react to small changes in pressure, the friction from a set of O-rings would prove too great. Thus, a tight clearance and notches were placed on this device as well.

The cap at the top of the cylinder allows access to the spool and provides a lip against which the upper spring can act. In the center of the cap is a hole which allows the ambient pressure to act on the spool. This hole also serves another purpose: when the spool moves, water must be forced through this hole. The hole is kept small, preventing quick movements by the spool, and the energy which is dissipated forcing the water through the hole prevents oscillation.

A slight overpressure in the displacement volume, which will help give the bellows a fuller shape, can be created by adjustment of the upper cap's position, Screwing the cap downward moves the spool, which is in the neutral position between the springs, further downward. From this point a greater pressure difference is needed to displace the spool far enough to stop air from flowing into the displacement volume. However, screwing the cap down too far will cause a pressure greater than the pressure relief valve can withstand, and air will be continuously vented through the system as one valve tries to increase the pressure and the other vents the bellows. Thus, care must be used adjusting the valves.

2.4 Springs

Using linear springs to counter the force upon the piston by the pressurized air allowed for a linear relationship between the pressure in the tanks and the position of the piston. This seemingly simple relationship became the driving factor in the design of much of the design.

It was decided to use the springs in a compression orientation because this avoided the problem of fixing the end of the springs which would arise if they were in tension. This orientation required a back plate on the device for the springs to push against. Six steel rods, four placed at the corners of the plates and two in the centers of the long sides, connected the back plate and the front plate. A third plated floated between the two, guided by the connecting rods, its position determined by the high pressure piston. The springs are attached to the back side of this floating plate, and the bellows are attached to the front side.

2.4.1 Spring Equations

Equations from Kempe's Engineers Year-Book [Kempe's 88] indicate the following for round wire helical compression springs:

$$\text{Rate} = Gd^4/8nD^3 \quad \text{pounds/inch} \qquad (1)$$

where G = modulus of rigidity (pound/in^2), d = diameter of wire (inches), n = number of active coils, and D = mean coil diameter (inches).

Similarly, the load P at deflection δ can be found using:

$$P = \delta Gd^4/8nd^3 \text{ pounds.} \qquad (2)$$

The stress on such a spring can be determined from the following equation:

$$\text{Stress} = 8PD/\pi d^3 \text{ pounds/in}^2. \qquad (3)$$

2.4.2 Spring Design

A piston diameter of 1 inch was assumed as a baseline case. However, when we used handbook equations to size the springs we found that the necessary spring was impossible: the wire diameter was greater than the radius of the coil. At this point the limitations the spring would put on the design became evident.

As the springs needed to be too large to counteract the load put upon them, the most obvious solution was to make the diameter of the piston smaller. This, too, proved to be insufficient. While the spring was possible to manufacture in this case, the collapsed size of the spring was still to large.

In addition to making the area of the piston smaller, a number of springs were used in parallel. This allowed each spring to bear a smaller portion of the load, and the load on each spring was reduced to tens of pounds.

Thus, it was possible to bring the total load on the springs down to about 400 pounds, and the handbook equations indicated that springs of this size were possible.

Using an assumed piston diameter, the maximum load on the springs could be calculated. In addition, a stroke of 7.75 inches had been derived from the layout of the displacement volume, so a rate could be approximated. Using the equation for stress to limit the maximum stress and making an assumption about the number of coils used in the spring provided two equations and two unknowns, d and D.

However, there is a grave error in this approach which lead to great misconceptions about the size of the springs. Talking to a spring manufacturer about the manufacturing process helped in understanding the proper interpretation and application of the equations.

During manufacture springs are specified for a certain rate, but due to uncontrollable variations in material characteristics it is difficult to get consistent springs. The variable which is controllable after the spring has been manufactured and tested is the number of coils. It is the number of coils which should be treated as a variable in using the spring equations.

2.4.3 Spring Characteristics

The springs used are catalog number 524 from Hardware Products Company, Inc. The outer diameter of the spring is 1.25 inches, and the wire diameter is .125 inches. The spring can be compressed from 12 inches to 4.1 inches, and has a spring rate of 8.7 lbs/inch. Solving equation 1 for n indicates that this spring has 32 coils, and at maximum load the springs will be stressed to 99000 psi according to equation 3.

The L/D ratio of these springs is very high and the springs must be supported to prevent them from buckling. To accomplish this the springs were run along the steel rods which connected the front and back plates, with one end of the springs grounded to the back plate, the other attached to the floating plate in the middle.

2.5 Materials

Devices which are designed for use in sea water must be very careful about the choice of materials. The conductivity of the salt water causes reactions between dissimilar metals which would otherwise happen at a negligible rate. In general, the rule of thumb is to not mix dissimilar metals.

In addition to this stipulation imposed by the sea, using dissimilar metals in piston- or bearing-type devices is also to be avoided. Unacceptably high wear is caused by the rubbing of dissimilar metals with different hardnesses. In both the piston and the air intake valve assemblies dissimilar metals would be self destructive.

The springs, being bought from a warehouse, are the only metallic parts whose material is already chosen. They are made out of steel piano wire, and all other parts of the device will be made out of steel as well.

Chapter 3

Final Sizing

The layout and sizes of the device can be seen in the three view drawing of figure 8.

Six springs with a spring rate of 8.7 lbs/inch, all of which will be compressed 7.75 inches, give a total load of 404 pounds. When the tank's pressure is 3300 psi, the piston should have a load of 404 pounds, so the area of the piston head is 404/3300 = 0.081 square inches. This corresponds to a diameter of 0.322 inches.

A piston wall which can withstand 3300 psi with a safety factor of 3 should be approximately 0.08 inches. However, because material is needed for fittings which will be attached to the cylinder, the walls will be made thicker, approximately 0.125 inches.

The piston shaft will bear all of the 404 pounds, and it is necessary to confirm that the shaft will not buckle under the load. Euler's equations for the critical load P_{cr} of a shaft is

$$P_{cr} = \pi EI/L_{eq}^2$$

where E is Young's Modulus, $I = \pi r^2/4$, and L_{eq} is the equivalent length, which, in the worst case scenario of high eccentricity, is 2L. This yielded a P_{cr} of 470 lbs, greater than the maximum load of 404 lbs. However, in order to increase this critical load to this value, it was necessary to increase the thickness of the bottom of the piston, which does not enter the cylinder, thereby decreasing the length of the thinnest section of the piston.

The steel bars which will attach the front and the back plates of the device must fit inside the ID of the springs. The OD of the springs is 1.25 and the wire thickness is 0.125, which means the ID is 1.25-(2*0.125) = 1.0 inches. Rods of 0.25 in diameter were chosen to act holding the two end plates together and provide lateral support to the springs. Since six of these rods will bear the 404 pound load, the resulting stress on the bars is 1400 lbs/in^2. This is well within the yield stress of the material, but also within the 6000 lbs/in^2 rule of thumb for stress at threads, which is important since the ends of these rods will be threaded for attaching nuts.

7.75 inches of bellows material compresses to approximately .5 inches. With .25 inch sections on the front and the floating plate for attaching the ends of the displacement volumes, this means that these two plates must not close within approximately 1.0 inch of each other. This is the neutral, zero force deflection of the springs.

The piston length must, of course, be at least the 7.75 inches which it must travel. In addition to that, it is advisable to have an overlap of at least six times the diameter, or 2.4 inches. Less than that and the piston will likely lock inside the cylinder walls, which will damage the mating of these two parts as well as causing the device to fail. Thus, the piston is expected to be withdrawn 11.2 inches into the cylinder.

Inside the cylinder a quarter inch of extra travel is provided, making the bore 11.45 inches long. 0.75 inches of extra material is provided for attachment of fittings from the high pressure lines, as well as a lip for bolting the part to the front plate and a small lip on the bottom face for centering the part.

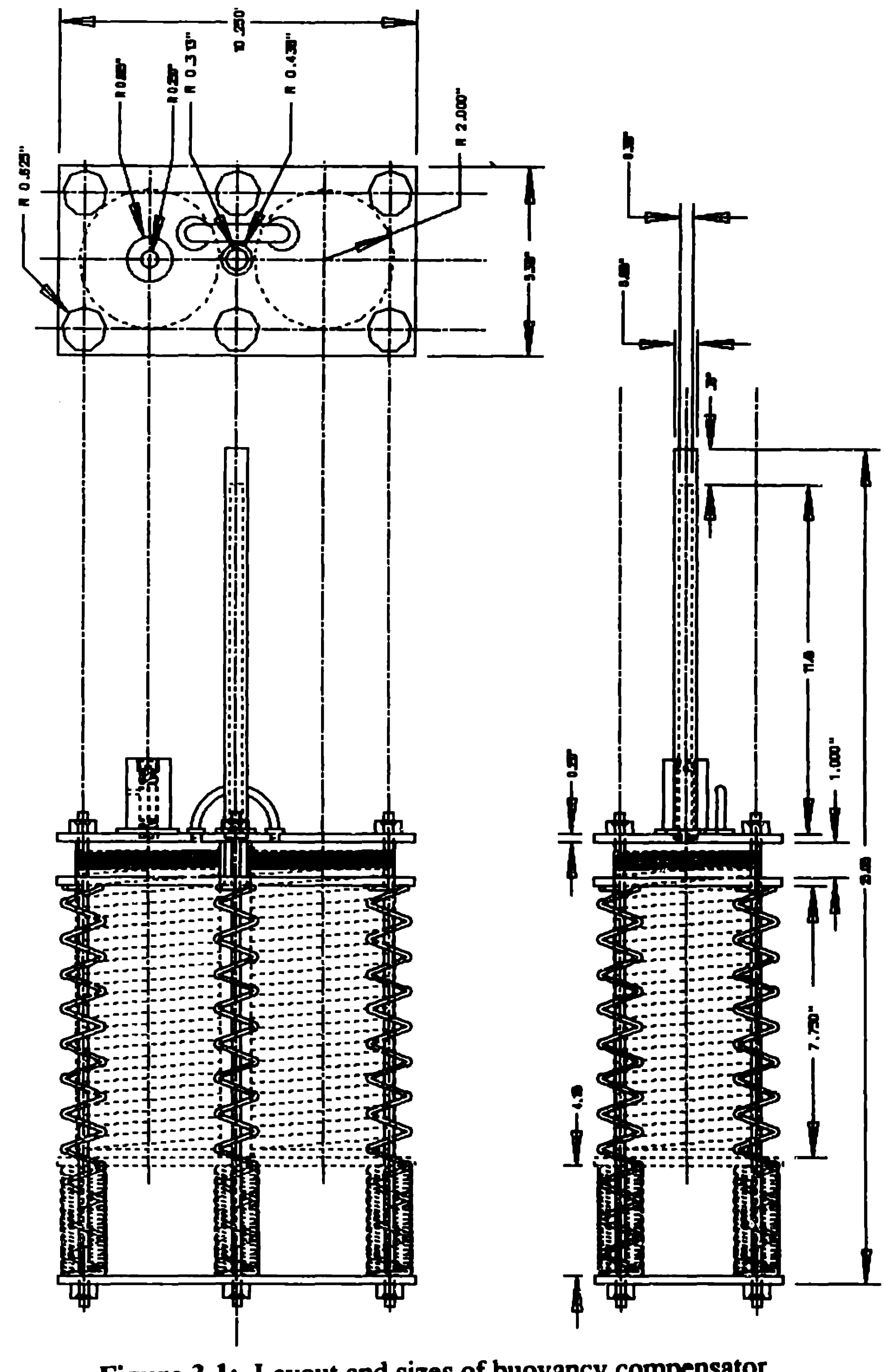

Figure 3-1: Layout and sizes of buoyancy compensator

Chapter 4

Use and Adjustments

The device was design for use with a ScubaPro 71.4 cubic foot tank, but some changes and adjustments can make the device compatible with tanks of a different size. In addition, these same adjustments will allow the device to be more finely tuned to the tank for which it is configured.

4.1 Displacement Volume Pressure Adjustments

The pressure inside the displacement volume will be regulated to a value close to the ambient pressure. However, the bellows can support some pressure difference, either negative or positive, between the pressure inside the bellows and the ambient water pressure.

A positive pressure difference would be desirable because it would help give the bellows the proper shape. The helical wire running the length of the bellows does provide shape; however, between the coils the vinyl tape has a tendency to sag. A slightly higher pressure inside the bellows would cause the bellows to balloon out, providing the proper shape and volume for the bellows.

This negative pressure difference can be achieved because the pressure release valve needs a non-zero positive pressure to vent the bellows. The air inlet valve can be adjusted so that the equilibrium pressure of the regulator is below this critical pressure of the release valve.

Screwing the top cap of the air intake valve downward increases the pressure which is needed inside the bellows to close the valve. Because this action moves the equilibrium point between the two springs downward, a greater pressure from the bottom, i.e., inside the bellows, is needed to move the spool up to its shutoff position.

Care must be taken that the shutoff pressure of the inlet valve is not greater than the critical pressure of the release valve. If this is the case, air will be vented continuously through the system, with one valve trying to increase the pressure inside the bellows and the other trying release that pressure.

4.2 Spring Rate Adjustments

The change in buoyancy of the device can be adjusted as well. In the case of either fine tuning the device or the use of a different size SCUBA tank, changing the rate of the springs will cause a change in the buoyancy characteristics of the device.

Should the device not be gaining enough weight to compensate for the use of air in the tanks, springs with a lower rate are necessary. This will cause both the stroke of the device and the volume of water displaced per unit of psi to be greater. Lower rate springs are obtained by using more coils on the spring. Since the use of more spring coils increases both the travel and the compressed length of the spring, the device must be longer to accommodate these lengths.

In the other, more dangerous case, just the opposite compensanatory acts are necessary. The spring rate is too low in this case, and reducing the number of coils will raise the rate. This will reduce both the stroke length and the compressed length of the spring, and thus the necessary overall length of the device.

4.3 Connecting to SCUBA Tank

Attaching the high pressure lines to this device must be done with care. Suddenly attaching a high pressure line to the device will cause the piston to drive the floating plate back. However, as the bellows can only be filled with air at a limited rate, a partial vacuum will be created in the bellows, causing them to collapse before the air inlet valve can adjust the pressure.

To avoid this problem, device should be attached to the high pressure lines before the valve at the SCUBA tank is turned on. After connecting the lines the valve can be slowly turned on, allowing the pressure inside the bellows to equalize during the expansion.

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