CONF-900720--2
UCRL- 102572
PREPRINT
Received by OSTI
JUL 0 5 1990

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This paper was prepared for submittal to
AIAA 26th Joint Propulsion Conference
Orlando, FL
July 16-18, 1990

June 8, 1990

Lawrence Livermore National Laboratory

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UCRL--102572

DE90 013472

Abstract

An aluminum piston tank has been developed for applications requiring lightweight, low cost, low pressure, positive-expulsion liquid storage. The 3 liter (183 in^3) vessel is made primarily from aluminum sheet, using production forming and joining operations. The development process relied mainly on pressurizing prototype parts and assemblies to failure, as the primary source of decision making information for driving the tank design toward its optimum minimum-mass configuration. Critical issues addressed by development testing included piston operation, strength of thin-walled formed shells, alloy choice, and joining the end cap to the seamless deep drawn can.

Introduction

Liquid storage tanks for aerospace applications typically operate under a variety of acceleration conditions, so special means are often necessary to ensure that liquid is always present at the tank outlet. For example, surface tension devices are one method to avoid having a bubble at the outlet in microgravity. More stringent needs in addition to the above are the prevention of liquid sloshing, and the avoidance of contact between the liquid and gas present in a vessel. In both cases, a free surface cannot be permitted, so the storage volume must decrease as the liquid is expelled. Flexible bladders and deformable diaphragms are commonly used to permit a changing volume while separating pressurant gas from the stored liquid.

The development effort described here was undertaken to fulfill a need for a lightweight, low cost, low pressure cylindrical tank which expels axially without sloshing.[1] The operational requirement for the vessel does not permit any positional shifting of the stored liquid, and the center of mass of the liquid must be maintained along the axis of the cylindrical tank at all times during expulsion. For this reason, a flexible elastomeric bladder could not be used alone to separate the liquid and pressurant. Deformable metal diaphragm technology is highly advanced for tanks with spherical and conical geometry,[2] but this technology must be modified for axially directed expulsion in a cylinder. A bonded rolling diaphragm is essentially a piston, with a metal diaphragm that peels off the tank wall to follow the moving piston.[3] A piston with a dynamic seal was the remaining choice for lower cost and lower mass, so this path was pursued in spite of possible leakage past the seal. An additional advantage of the chosen concept over a bonded rolling diaphragm is reusability.

Development goals included a liquid capacity of 3 liters (183 in^3) with minimal ullage volume, a burst pressure greater than 1.4×10^6 N/m^2 (200 lb/in^2), and a tank weight less than one-tenth the weight of stored water. The performance goal can be summarized concisely as $pv/w > 1400$ meters (55,400 inches), where p is the burst pressure, v is the volume of stored liquid, and w is the weight of a complete tank. Although this number may be less than impressive when compared to the state of the art in pressure vessel design, it should be noted that the low pressure and low cost requirements precluded the use of materials with higher specific strengths than aluminum. In particular, a comparable titanium tank would have extremely thin walls and could not be deep drawn. A composite tank would still require a thin leaktight metal liner on which to wrap the fibers.

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To achieve high manufacturing yields and reliability, the development effort emphasized the use of manufacturing methods which inherently permit a high degree of consistency within a lot of parts or assemblies. In addition to developing a working tank, an equally important goal has been to demonstrate repeatability at each manufacturing step, to permit replacing individual inspections and acceptance testing with lot acceptance testing. Deep drawing, ironing, hydroforming, cnc machining, dip brazing, and automated electron beam welding are able to satisfy the above criteria.

The development process relied upon extensive destructive testing of piece parts and assemblies almost exclusively for objectively evaluating designs, materials, and manufacturing methods. Hand stress calculations determined basic dimensions. The value of more detailed calculations was limited by unknowns, particularly material properties in the as-formed and as-welded conditions. Finite-element analysis was undertaken, but the results had little influence on the final design. It is, however, noteworthy that the method of design evolution through feedback from destructive testing was particularly applicable to this project due to the low cost of test specimens, made possible by their small size and by the fact that mass production methods were used from the outset.

During development, minimal effort was devoted to estimating time to completion, or to keeping assembly drawings predicting the final design. Rather, numerous possible designs were pursued in parallel, and selection was based on test data. Every idea which had been conceived or tried was documented in a periodically updated comprehensive report which was organized according to piece parts and joining methods, and which contained a synopsis of all test data. Frequent reviews of this report helped guide the team along the path of least total effort required to reach the goal.

Tank Design

Figure 1 shows the design of a complete tank, which consists of a deep drawn can containing a piston, with a cap welded on. Liquid is stored between the piston and the cap, so the piston is dished to mate with the cap for high expulsion efficiency. During expulsion, a pressurant resides in the ullage volume between the piston and the domed end of the can. The cap is concave as viewed from the outside of the tank, which facilitates packaging of associated hardware that must be structurally attached to the cap flange. The concave cap also permits the most compact piston envelope, while minimizing both ullage volume and overall tank length. Alternatively, a convex cap would permit fiber overwrapping of the tank if this were desired for operation at higher pressures.

The piston is nominally intended to operate with a wet seal, i.e. without an auxilliary bladder to separate the pressurant from the stored liquid. The possibility of small amounts of the two fluids mixing limits the range of applications for this particular design. Reactivity, cross-solubility, and contamination were considered for intended applications, and were not found to be of great concern.

The tank is all-aluminum, except for polytetrafluoroethylene piston seals which are loaded with stainless steel springs. Aluminum was chosen for its low cost, low density, cold formability, and machinability. Strength, weldability, and compatability with stored liquids influenced alloy selection, as is explained further in the section describing can development.

Aluminum was also chosen for its high heat conductivity. The tank is intended to be used with a pressurant which is in

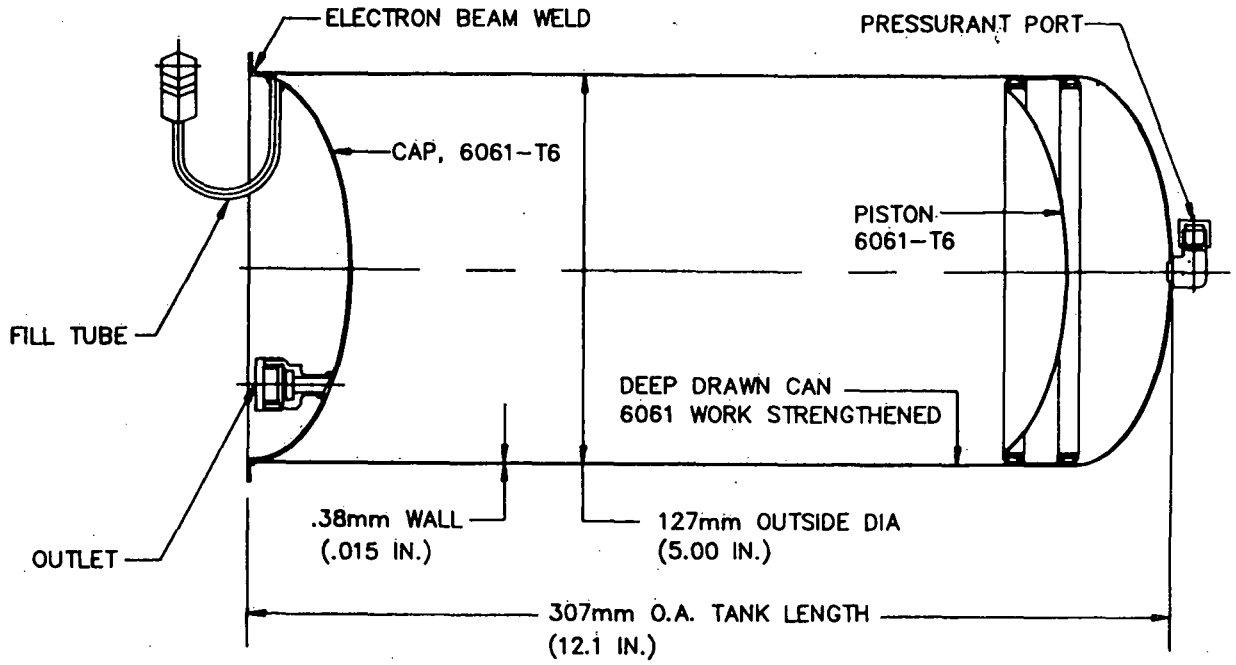

Figure 1. Design layout of the lightweight aluminum piston tank.

liquid-vapor equilibrium, so heat input is required to evaporate the pressurant as the ullage volume increases during expulsion. Heat conduction through the face of the piston and in the wall of the tank keeps the pressurant warm.

Test Methods

The basic tool for failure testing of tanks and piece parts was a benchtop hydrostatic pressurization apparatus consisting of a stainless steel hand pump for water (Enerpac 11-100), the outlet of which terminated within a transparent safety enclosure fabricated from half inch polycarbonate plate bolted to an extruded aluminum frame. Leaking, yielding, buckling, and bursting could thus be directly observed visually. The design of the enclosure was based on calculations of energy released upon vessel bursting, and impact tests. The polycarbonate withstood many times the maximum theoretical fragment momentum. This approach obviated the need for a reinforced concrete test cell, for example, as is traditional for experimental work with pressurized systems.

The vast majority of failure testing during development was done on piece parts rather than on complete tank assemblies. For example, pressurization fixtures using o-ring face seals were used for testing cans and caps independently. In the case of the cap, it has a flange as designed. The can is unflanged for the tank design shown in Figure 1, but a flange was formed on many cans from each manufacturing lot to facilitate testing of can strength and pressurant fitting welds, in addition to candidate cap-can joints which required a flange on the can.

Non-structural development testing included operational testing of pistons and tanks, also using water. Several of these experiments are described below.

Piston Development

The first part to be tested was the piston itself, since piston tanks are relatively rare and the feasibility of a lightweight piston had to be proven. As is shown in Figure 2, a separated pair of spring-loaded seals, specifically designed for reciprocating applications, was chosen. The most important initial question was what piston length (i.e. longitudinal distance between the seals) is needed to prevent cocking. Piston length is critical in determining tank performance, since its product with cross sectional area trades off directly with storable liquid volume within a given cylindrical vessel. Sources of cocking moment include longitudinal friction which is unevenly distributed around the seal perimeter, and a hydrostatic pressure gradient due to transverse gravity or acceleration.

Piston cocking

Using the y-axis and constants defined in Figure 2, the cocking moment due to a transverse pressure gradient within a liquid having a given density, ρ, can be calculated.

$$\text{Cocking moment} = \int p\, y\, dA = \rho\, a \int y^2\, dA = \rho\, a \frac{\pi}{64} D^4,$$

where the integration over the piston area, A, has been performed by substituting the familiar expression for the area moment of inertia of a circular cross section. Assuming water's density, and a .127 meter (5 inch) diameter bore, the cocking moment is 0.13 N-m (1.1 in-lb) if the transverse acceleration, a, equals earth's gravity.

The cocking moment must be reacted by radially directed seal forces applied by the wall of the bore, so seal springs with a high radial stiffness were chosen. One test of the effect of cocking moment consisted of running a heavyweight solid piston in a horizontal transparent plastic bore. A steel bar extending axially from one side of the piston applied a 0.75 N-m (6.6 in-lb)

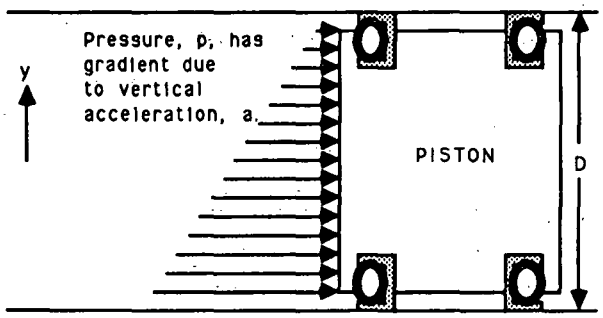

Figure 2. Piston cocking is prevented by having two seals. One source of cocking moment is the pressure gradient due to transverse acceleration.

cocking moment due to gravity, and the piston ran the length of the bore without mishap.

In a heavyweight apparatus and in prototype tanks, the pressure drop across the moving piston has been measured at approximately $2x10^4$ N/m^2 (3 lb/in^2). The total running friction on the piston is therefore roughly 250 N (60 lb), so an uneven distribution of this friction could cause a far greater cocking moment than the pressure gradient effect. In particular, if half of the frictional force is applied at one point along the seal circumference with the remainder of the force uniformly distributed, the cocking moment would be 8 N-m (75 in-lb).

In one test, various cocking moments were applied to a heavyweight piston in a deep drawn can which was slightly pressurized. Approximately 11 N-m (100 in-lb) was necessary to open a leak path past the seals, a condition which occurred at a cocking angle of .07 radians (4 degrees). Given the calculations above, this result indicates that only a severe defect would result in a cocking moment large enough to open a leak path. Piston cocking due to unevenly distributed friction around the seal perimeter has been observed, but was found to be due either to piston damage, or adhesion of the piston seals to the tank wall associated with corrosion and mineral deposits after long term tests with tap water.

Seal performance

Initial leak tests of the piston seals consisted of a long-term exposure to water loaded by gravity, with the tank vertical and 0.1 m (4 in) of water on top of the piston. Under this condition, leakage was less than 1 cc/month over three months. To improve upon this, other work is in progress, including a test in which the void between the piston and the tank wall, bounded by the two seals, will be monitored to aid in understanding leakage.

Another factor influencing leakage past the piston is the presence of particulate contamination, particularly if a high piston cycle life is desired. Complete testing of leakage v.s. piston cycle life awaits an assembly which has been subjected to high-purity cleaning.

Lightweighting

The final aspect of piston development was mass reduction. Using 6061-T6 on a numerically controlled lathe, along with a low melting temperature metal for temporary support, the elliptical piston face as shown in Figure 1 has been fabricated as thin as 0.25 mm (.010 inch), resulting in a piston mass of 43 grams, almost half of which is the seals. One critical structural test was the application of differential pressure to the lightweight piston when constrained by the domed end of the can as shown in Figure 1. It passed a test of the filling pressure requirement, set at 10^5 N/m^2 (15 lb/in^2), with slight deformation noted at $2.7x10^5$ N/m^2 (40 lb/in^2).

Can Development

After it became apparent that a cylindrical tank would be feasible due to preliminary tests with pistons and seals in a heavyweight bore, attention was devoted to the main body of the tank. A basic familiarity with aluminum forming processes and alloy characteristics was obtained from a manufacturer's reference[4] after contemplation of the low cost, high volume production of beverage cans. Deep drawing is used to form the basic seamless shape from a flat sheet blank, as indicated in Figure 3. Ironing, a process in which the part is pushed through a tool ring smaller than the part, Figure 4, then reduces the sidewall thickness while strengthening the aluminum. The challenge for the present work was to form cans much larger than beverage cans, using a high strength, weldable alloy. It was soon discovered that beverage can manufacturers did not have presses large enough, and forming shops having the necessary equipment were unfamiliar with the details of manufacturing thin aluminum cans. Fortunately, one of these shops was willing to learn by trial and error.[5]

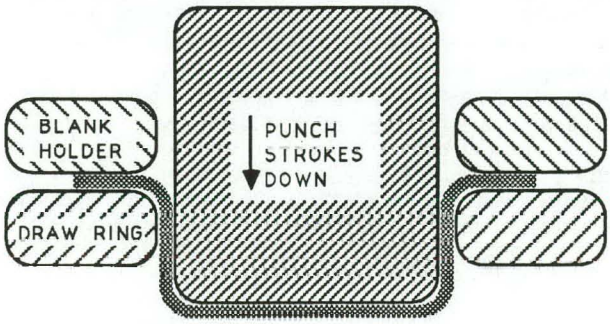

Figure 3. The deep drawing process forms a can from a flat circular blank.

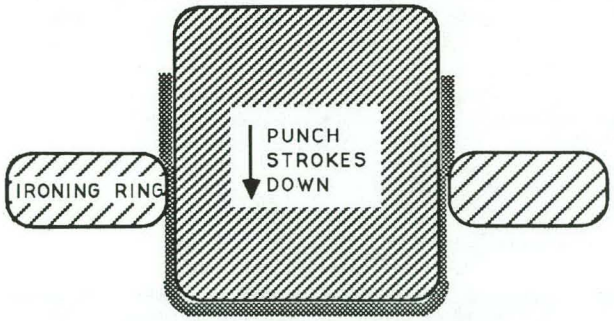

Figure 4. The ironing process reduces the sidewall thickness while stretching the can.

6061 cans

It was reasoned that a 6061 can would provide the greatest strength among alloys which were known to be cold-formable and readily available. After tooling and process development, drawn and ironed cans having a sidewall thickness of .38 mm (.015 in) were obtained from 6061-T0 sheet. Some of these cans were flanged at the open end, to permit burst testing using an o-ring face seal and flange clamp. The rupture is a longitudinal split several inches long, resulting from hoop stress, as can be seen in Figure 5. The yield and ultimate hoop stresses of the as-formed 6061 aluminum cans are nearly the same, at approximately $2.3x10^8$ N/m^2 (34,000 lb/in^2). This value is calculated from a nominal burst pressure of $1.4x10^6$ N/m^2 (205 lb/in^2) which has been consistent within 3 percent over several manufactured lots.

The 6061 alloy is not normally used in a work-hardened condition. The T6 heat treated material is known to be capable of an ultimate strength of $3.1x10^8$ N/m^2 (45,000 lb/in^2). In an attempt to achieve this increased strength, cans were solution heat treated and quenched by several methods. Quenching in water resulted in a dimpled, wrinkled can wall, but the expected material properties were obtained, with an ultimate strength above $3.2x10^8$ N/m^2 (46,000 lb/in^2) based on hoop burst data.

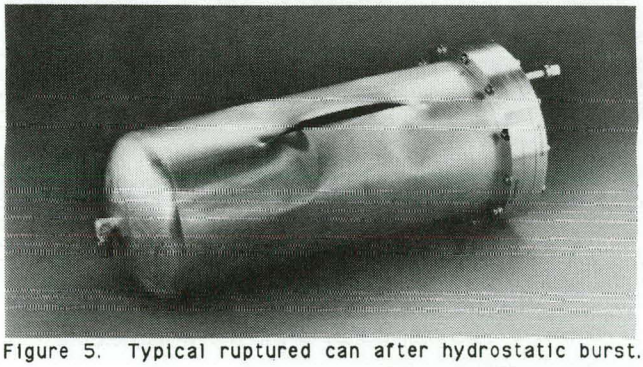

Figure 5. Typical ruptured can after hydrostatic burst.

Several cans were quenched over a mandrel in water, but this required a tight fit to avoid wrinkles, resulting in stretching of the can because the thicker mandrel cooled more slowly. Two cans that were quenched in air without a mandrel did not have wrinkles, but there was a slight overall distortion. The best result was obtained by quenching in a water mist both inside and outside the can, resulting in minimal dimensional differences from the as-formed part.

Other alloys tested

Unfortunately, welding 6061-T6 weakens it considerably over a heat affected zone near the weld, and heat treating of complete tanks is unacceptable due to the elastomeric seals inside. This is not an insurmountable limitation, as explained below regarding cap-can joining, but other alloys have been under consideration. For example, the recently developed aluminum-lithium alloy[6] is easily weldable and is very strong in the as-welded condition. For these reasons, an attempt was made to form cans of this material, but drawing it was unsuccessful, apparently due to rapid work-hardening.

After developing the 6061 aluminum cans, a surprising result was obtained with beverage cans, which are made of 3000 series aluminum alloys. It had been assumed at the outset that 6061 had the potential for greater strength than the standard can alloy which is not heat treatable. However, two beverage cans which were burst with the ends restrained were found to have an ultimate hoop strength of 3.3×10^8 N/m^2 (48,000 lb/in^2), as calculated from a wall thickness of 1.0×10^{-4} m (.0040 in), a radius of .033 m (1.3 in), and a burst pressure of 1.01×10^6 N/m^2 (147 lb/in^2). Clearly, this strength is due to the extreme amount of deformation during forming, particularly ironing.

Design of drawn and ironed parts

It was learned through experience on the present project that the manufacturing methods used and the design details are highly interdependent, as is the case with beverage cans. One early mistake was the assumption that the bottom of the can would have the same thickness as the cylindrical sidewall. This led to a request for a hemispherical can end to guarantee burst failure in the tank sidewall, given that stress in a spherical pressure vessel is half the hoop stress in a cylinder of the same radius. This approach seemed reasonable after experience with short hydroformed cans which were made very early in development with a hemispherical dome.

However, a hemispherical end was found to be incompatible with the draw-and-iron method, for two reasons. Parts with flat ends are the easiest to deep draw, and drawing a hemisphere was attempted without success. Secondly, it is desirable to start with aluminum sheet more than twice as thick as the final sidewall thickness, so that the required ironing is enough to significantly strengthen the material. Since ironing affects only the cylindrical part of the can, the dome end will be thick enough to provide adequate strength with a relatively flat profile.

Another fact learned about the draw-and-iron manufacturing method is that it is possible to form cans having significantly greater length than in the present design. Ironing lengthens the can as the wall is thinned. The prototype drawing operations formed a can nearly as long as the final part, so a significant amount of metal had to be trimmed after each ironing step. Attempts to iron the walls thinner than .38 mm (.015 in) resulted in significant longitudinal scratches, which suggested that the 6061 material had been strengthened enough to prohibit further working.

Cap Development

Forming the caps does not require deep drawing, since the depth of the draw is minimal. Instead, the parts are hydroformed, a flexible die forming process in which a rubber diaphragm with hydraulic pressure above it is used to form the sheet metal over the punch as the punch rises, as indicated in

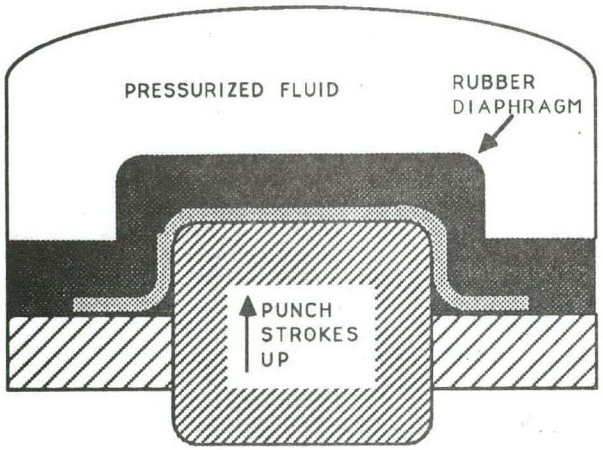

Figure 6. In hydroforming, hydraulic pressure forces the sheet to conform to the punch shape.

Figure 6. The strength of the resulting caps is due to the 6061-T0 starting material, since deformation is minimal. However, the caps are easily heat treated without warping, since they don't have a large cylindrical surface and the thickness is twice that of the can sidewall.

Cap buckling

In a pressurized tank, the 2:1 ellipsoidal cap shown in Figure 1 is compressively loaded since its center of curvature is outside the tank. Therefore the structural requirement for the cap was to avoid buckling. Modeling buckling of the concave cap using the finite element method was challenging because of the sharp discontinuity at the boundary. The first prediction was that buckling pressure would be 1.7×10^6 N/m^2 (250 lb/in^2) for a cap 0.5 mm (.020 in) thick. Actual parts dimpled at the center at roughly 0.6×10^6 N/m^2 (90 lb/in^2). It was then understood that due to the sensitivity to slight geometric imperfections, buckling pressure may only be determined by fabricating and testing parts. Depending on the assumed size of the initial imperfection, the finite element model could be made to predict buckling at any pressure up to that given above for a perfect shell.

Increased thicknesses that were tested depended mainly on rapid availability of 6061-T0 sheet. With a thickness of 0.97 mm (.038 in), the initial dimple occured off-center at roughly 1×10^6 N/m^2 (150 lb/in^2) without heat treating. With heat treating, these parts held to the maximum test pressure of 2.3×10^6 N/m^2 (340 lb/in^2), indicating that thinner caps would be acceptable for reduced weight. After heat-treating parts made of thinner 0.81 mm (.032 in) material, the buckling failure was an off-center dimple at 2.25×10^6 N/m^2 (326 lb/in^2), repeatable to within 2 percent.

An alternate alloy was tried, in an attempt to facilitate welding the cap to the can and welding the tubes to the cap. It was reasoned that if the cap could be made of the alloy used as filler for 6061 welds, separate pieces of filler material would be unnecessary. Several caps were formed from 0.76 mm (.030 in) thick 4047 aluminum (Alcoa 718, 2.46×10^8 N/m^2, 35,600 lb/in^2 ultimate), but they buckled below 1×10^6 N/m^2 (150 lb/in^2). Caps were also successfully formed from Al-Li alloy.[6]

The 0.81 mm (.032 in) thick 6061-T6 cap design was retained for development of complete tanks, although further thinning could result in useful mass reduction. Doing so successfully would require a better understanding of the buckling problem. Paradoxically, sensitivity to imperfections as determined from the model suggests elastic buckling, whereas the observed dependence on material strength indicates plastic buckling. Some caps have had reduced buckling pressures related to severe bending of the outlet tubes. Therefore, further reduction of the cap thickness could require strict avoidance of bending the tubes during handling.

Tube joining by dip brazing

As is shown in Figure 1, the liquid fill tube and the tank outlet are joined to holes in the cap. In order to avoid the cost of individual welds, a dip brazing process was pursued for these joints. The caps are pierced and swaged out to form conical bosses into which the fill tube and outlet fitting tube fit snugly. After 4047 aluminum filler is added, the parts are submerged into a temperature-controlled molten salt bath. After dip brazing, the cap braze assemblies are solution heat treated, quenched, and aged to the T6 condition.

The dip braze joint strength is enhanced by the relatively large amount of material added, which essentially forms a fillet along with the conical boss formed as part of the cap. The acceptance criterion for joint development was the capability to bend the tubing without affecting the braze.

Independently from working on the dip brazing process and cap thickness optimization, numerous caps without holes or tubes were made for testing tank assembly methods. Caps were formed with approximately five different edge geometries, to support testing a number of different cap-can joint designs.

Joining Cap to Can

The low cost of formed cans ($26) and caps rendered testing of cap-can joints quite inexpensive. It was not necessary for tubes or fittings to be welded or brazed to either part. Instead, a Schrader valve normally used for tubeless motorcycle tires was installed through a hole drilled in the dome end of each can. Pressurizing the cap-can joints to failure required hoop restraint over the entire length of the can, which was accomplished by using a piece of large plastic pipe. The criterion for joint design acceptance was to repeatably demonstrate a failure pressure significantly greater than the nominal burst pressure of the cans, without leaks.

The weld problem

The final cap-can joining method and joint configuration, Figure 7, was uncertain for a long time, due to two fundamental aspects of tank manufacture. Heat treating the tanks after welding the cap on is unacceptable, due to the nonmetallic piston seals in the tank, the high likelihood of distorting the thin aluminum sections, and the need to prevent contamination of the internal surfaces by oxidation. Compounding this problem is the fact that the ironing process prevents forming a thickened weld preparation ring on the open edge of the can, since the part must be pushed completely through the ironing ring in order to remove it from the tool post. These considerations initially suggested that a welded joint would be suboptimal, since material strength is lost during welding.

Given that axial stress in a thin-walled cylindrical pressure vessel is half of the hoop stress, an optimum cap-can weld requires an as-welded material strength greater than half that of the ultimate hoop strength of the can. A reference value[4] for the ultimate stress of 6061-T0 aluminum (1.24×10^8 N/m^2, 18,000 lb/in^2) is less than half that of 6061-T6, but greater than half of the as-formed can's hoop strength. Although it is a compromise from maximum possible material strength, the welded tank using the as-formed can, shown in Figure 1, is nevertheless a logical design because the weld outperforms the strength of the vessel as a whole.

Weld development

To quickly confirm that the achievable as-welded material strength could be greater than half of the ultimate hoop stress of the can, GTA welds were tested initially. Axial strength of the weld was adequate, but there was a significant bulge due to hoop yielding in the heat affected zone, a 2 cm (0.8 in) wide band around the can adjacent to the weld. Note that the weld bead itself was not particularly weak, since there was excess material there. However, the heat-affected unmelted material near the weld contains both solutioned material and overaged material, both of which have been weakened due to diffusion of alloying elements.[7]

The yield stress of 6061-T0 is only a small fraction of its ultimate stress, so it was apparent that a successful weld must have a heat affected zone so small that any yielding which occurs does not permit significant dimensional changes in the tank. It is well-known that the heat input can be greatly reduced by welding with a laser beam or electron beam. Both of these methods were pursued in parallel, with several different joint configurations and filler metal dimensions. Both methods produced welds of adequate strength, but the laser welds often developed small leaks during hydrostatic pressurization.

Final selection of the e-beam corner weld shown in Figure 7 was based on testing a lot of seven cap-can weldments. With the exception of one weld which could not be tested to failure due to cap buckling, all burst above 1.5×10^6 N/m^2 (220 lb/in^2) with hoop restraint, and the rupture point was in the heat affected zone within a few mm of the weld. Two of the welds developed leaks at a low enough pressure that they would be eliminated by a leak check at proof pressure.

Electroplated nickel joint

When the tank design was originally conceived, welding was ruled out. Separately from the weld strength concerns discussed above, there appeared to be a need for a bimetallic joint due to a design requirement for a stainless steel or titanium cap. Some years earlier, a colleague had developed a method of joining dissimilar metals using electroplating.[8] After it became apparent that the cap would be aluminum, this joining method was kept under parallel development because it would permit taking full advantage of the strength of heat treated cans. Work on the plated joint was stopped when the electron beam welding capability was proven and it became apparent that the compromise in tank strength was acceptable.

Figure 8 shows the nickel plated joint. The cap is designed so that the axial tension in the can is carried into the cap by a band of nickel plated around the tank. After appropriate

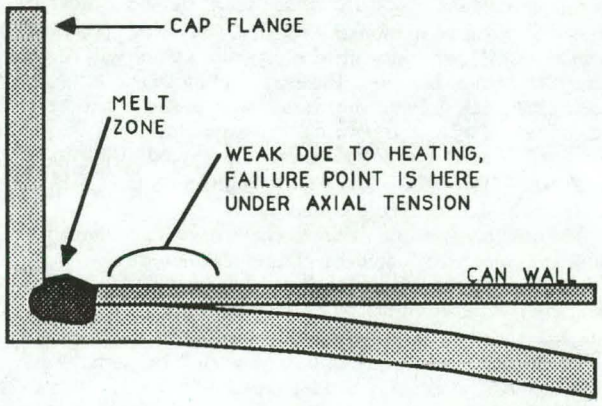

Figure 7. The electron beam cap to can corner weld.

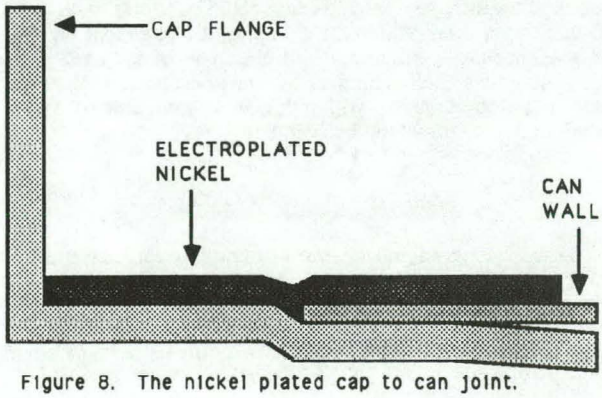

Figure 8. The nickel plated cap to can joint.

cleaning, masking, and final surface preparation, the tank assembly is held vertically and rotated in the plating solution, with a deposition rate of 0.3 mm/day (.012 in/day). This joining method does not weaken the aluminum, because all processing steps are carried out near room temperature. The strength of the joint is limited by the thickness of the nickel, and by the product of area and shear strength of the aluminum-nickel interface.

The groove at the center of the nickel band was not intentional, but was due to the fact that the buildup of plated nickel conformed to the underlying surface geometry. Pressurizing a tank with the ideal plated joint would put the nickel in uniform tension, but the groove resulted in bending stresses. Extra nickel thickness was required to overcome this imperfection. Another problem encountered during development testing was delamination at the aluminum-nickel interface. This problem was certainly process-dependent.

Of approximately fifteen development cap-can assemblies joined by nickel plating, half did not have enough nickel, and some failed due to difficulties in maintaining consistent processing conditions. However, four were pressurized (with hoop restraint) significantly above 2.1×10^6 N/m^2 (300 lb/in^2) without structural failure. In one case, the dome end of the can ruptured at 2.9×10^6 N/m^2 (420 lb/in^2). These successful joints each had a nickel band approximately 0.5 mm (.020 in) thick and 13 mm (0.5 in) wide, which adds almost 20 grams to the tank mass.

Although the electroplated nickel joint was demonstrated to withstand higher pressures than heat treated cans, several problems had not been solved at the time work was stopped. Preventing processing solutions from seeping into the tank had not been seriously addressed, but this would have to be done to keep the tanks clean inside. If gas pressure is allowed to build up inside the tank, then bubbles escape through the joint, preventing leak paths from being sealed with nickel. Some assemblies were vented at the pressurant end, and some were sealed, but both kinds developed joint leaks below the structural failure pressure, and most had visible voids in the groove.

The electroplated nickel joining method is certainly more applicable to unpressurized structures where sealing against leakage is not an issue. If the leakage and cleanliness problems had been overcome, the design tradeoff relative to the electron beam welded tank would depend on whether the additional mass of nickel and additional tank length due to the joint geometry are significant disadvantages.

Welding caps to heat treated cans

Although the nickel plated joint now appears unattractive for reasons other than strength, the possibility of deriving benefit from the increased strength of heat treated cans has not been ruled out. Re-aging after welding may strengthen the weld, if the weakest part is the solutioned material rather than the overaged material.[7] The aging temperature would be cool enough to avoid damaging the piston seals. Note that slight weakening of the heat treated can would be acceptable, as long as the final weld strength is greater than half of the can's hoop strength.

Perhaps a more brute-force approach to taking advantage of the strength of fully heat treated cans would be to start with extra wall thickness, then reduce the thickness by chemical etching after masking the weld area at the open end of the can. This could result in a tank which is lighter and stronger than the presently preferred design, with a thickened weld area so the as-welded joint strength would be adequate.

Operational Testing of Assemblies

To enable operational testing of complete tank assemblies before the cap-can joint problem was solved, a heavyweight flange with o-ring seals, as shown in Figure 5, was used to attach the cap to a flanged can. In addition to permitting tank testing in parallel with joint development, this bolted tank could be easily disassembled for inspection after a test. Furthermore, installing a piston into a bolted tank's flanged can is considerably easier than installing a piston into the straight-ended cans required for welded tanks.

Several bolted tank tests were run at Rocket Research Company, including an expulsion test with a propane-isobutane mixture in liquid-vapor equilibrium. It was demonstrated that thermal equilibrium could be maintained, resulting in a constant tank pressure during rapid expulsion. Further testing included numerous piston cycles, which resulted in an enhanced appreciation of the importance of high purity cleaning and interior surface finish toward ensuring a high piston cycle life. High purity cleaning, which is presently being pursued, requires the direct impingement of high velocity liquid jets to all surfaces before assembly, in order to dislodge particulate matter.

Several desired operational modes and operational restrictions were identified over the course of working with bolted tank assemblies. It is desirable to evacuate the liquid side of the tank just prior to filling. However, the cans collapse if internal pressure is 6×10^4 N/m^2 (9 lb/in^2) or more below ambient external pressure. Therefore, the piston must be drawn fully against the cap before evacuating through the liquid fill tube. Also, the vacuum may be applied for only a limited time if the pressurant port is sealed, given nonzero leakage past the piston seals. During filling, the liquid pressure must be limited to about 1×10^5 N/m^2 (15 lb/in^2) to avoid crushing the piston against the pressurant end of the tank. After filling, there must be some margin left for piston motion which is driven by thermal expansion of the liquid.

Safety and Reliability Considerations

Pressure vessel safety considerations are twofold. If a vessel bursts, i.e. fails catastrophically, its stored energy and stored fluid are released in a very short time, in which case the greatest hazard is typically the energy release. If a vessel leaks, the energy and fluid are released over a longer period of time, in which case the energy release hazard is usually not of concern. Leakage may be insignificant or hazardous, depending on the toxicity of the stored fluid.

The low working pressure and volume of the aluminum piston tank is such that the energy release upon bursting is only a minor hazard. The United States Air Force pressure safety standard does not even consider a small tank operating below 6.9×10^5 N/m^2 (100 lb/in^2) to be a pressure vessel, unless leakage would be a danger due to the nature of the stored fluid.[9]

If the aluminum piston tank is used to store a toxic liquid, bursting would present a greater hazard than leaking, due to the rapidity of fluid release in the former case. Therefore, it is still useful to distinguish between leaking and bursting failures, which are independent phenomena. For the present tank design, leakage is always due to a local flaw in either a weld or in a formed part. Bursting only occurs when the hoop stress reaches the ultimate strength of the material over a wide area.

For aerospace pressure vessels, a "leak-before-burst" criterion[9] requires demonstration that any initial flaw which propagates through the wall of a pressure vessel will result in leakage instead of bursting. Pressure cycling causes fatigue and accelerates crack growth, so it is desirable to show that pressure cycling at or above operating pressure does not result in bursting. One of the bolted tanks was cycled 1000 times to 1×10^6 N/m^2 (150 lb/in^2), and remained intact. Subsequent burst pressure was 5 percent below nominal.

The effects of several different flaws have been investigated. Burst pressure is affected by long scratches due to ironing, which are not merely local flaws since the effective wall thickness is reduced over a large fraction of the can's length. All cans have visible longitudinal lines approximately 6×10^{-6} m (.0002 in) deep, which affect burst pressure by perhaps a few percent. Any can with a deep scratch must of course be discarded.

The 6061 material which has been used for can development has been noted to contain some inclusions of foreign particles. Among the 30 or so cans represented by burst data to date, one had numerous inclusions approximately 1 mm in size, but burst pressure was nominal and the burst opening was not near any of the defects which had been marked before the test. Most cans do not have such obvious, large defects, and there have been no anomalously low burst pressures throughout development testing. The data includes one can with corrosion pits approximately 0.3 mm in size, which also did not affect the burst pressure or location.

To more dramatically demonstrate that local defects are not expected to reduce burst pressure, a bolted tank was pressurized with water to approximately 1×10^6 N/m^2 (150 lb/in^2), or about 3/4 of its burst pressure. A nail was driven into the can sidewall, and a stream of water exited the hole, but there was no rupture propagation as is characteristic of bursting.

The primary purpose of operating the tank significantly below ultimate hoop stress is to ensure that bursting will not occur. A significant random sample of cans from a mass-produced lot are burst to verify the expected pressure value. For tanks which are to be put into service, a proof pressure test is then performed between operating pressure and nominal burst pressure. Since bursting at proof pressure is highly unlikely, the primary purpose of the proof test is to selectively eliminate tanks which have flaws that may cause leaks during operation.

Discussion

The development of a working lightweight piston tank is essentially complete, and transfer of the technology to industrial affiliates is underway. The most important issue remaining to be addressed is integrity of the piston sealing capability over long lifetimes, which may require improvements in the seals and the interior can finish. For some applications using a liquid-vapor pressurant, a surface tension device may be needed behind the piston to prevent pressurant slosh.

A number of performance improvements appear to be possible, including reducing the thickness of the tank ends, as well as reducing sidewall thickness if material strength or weld strength can be improved. If heat treated cans are used, for example, the strength of the as-formed can would not be of concern so less ironing would be needed. In this case, it may be desirable to begin with a thinner blank, which would be consistent with both a thinner can end and fewer required ironing steps, thus reducing cost as well as mass. A more significant cost reduction would result from developing a formed piston to replace the machined piston.

The methods used for the project described herein are generally applicable to engineering development. It is unproductive to be overly concerned with the final detailed design if there are numerous unknowns. It is far more productive to spend time eliminating the unknowns. In particular, design decisions should be made after obtaining objective test results whenever possible. New elements of a design should be tested as independently as possible to reduce development time and to avoid expensive failures. Finally, it is important to learn as much as possible from each mistake, so it is worth documenting the details of failures as well as successes. Essentially, this is an engineering application of the well-known scientific method.

Acknowledgments

The following individuals made contributions to the effort and/or contributed design suggestions: Gerry Brewster, Tod Dalrymple, Roger Decker, Jack Dini, Anna Ekshtut, Ken Fisher, Stuart Krupoff, Debbie Leal, Frank Lego, Jim Luddy, Ron Lustig, Jeff Maybee, Bob Pasco, Kent Pugmire, Jeff Rozek, Tom Schafenacker, Walter Scott, Chris Stephani, George Sutton, Don Swink, Chet Westlock, and Lowell Wood.

This work was sponsored by the U.S. Government and performed by the University of California Lawrence Livermore National Laboratory under Contract W-7405-Eng-48 with the U.S. Department of Energy.

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