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APPLICATION OF PHOTOELASTIC COATINGS TO RING-TYPE COMPRESSOR VALVES

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ABSTRACT

This paper discusses the use of photoelastic coatings to measure flexural stresses in very thin flexural specimens (0.008 inches thick refrigeration compressor valves). A suggested procedure for applying the coatings to thin specimens having intricate contours is described in detail; the method involves the use of pre-cast plastic sheets. Correction factors to account for the influence of the plastic and cement on measured stresses are discussed. A theoretical correction factor is developed for equal load and equal deflection conditions; use of this correction factor is shown to yield accurate relative stresses. An experiment conducted on specimens having a geometry similar to the valve geometries and having known stress concentration factors shows that an additional experimental correction factor must be used to obtain absolute stresses in thin specimens.

INTRODUCTION

This paper is an outgrowth of the desire to measure flexural stresses in ring type compressor valves in order to predict their fatigue lives. It was desired to predict the relative fatigue lives of the two valves shown in Figure 1. The valves shown are approximately two inches in diameter, 0.008 inches thick, and are made from spring steel. Cross-hatched areas represent clamped portions. The two tabs on each valve which are not clamped operate against stops which limit valve deflection.

Relative fatigue life predictions are dependent upon dynamic stress levels. Dynamic valve stress measurements using miniature epoxy-backed foil strain gages in operating compressors have been reported by previous investigators of cantilever type valves where it is known that the peak flexural stress normally occurs at the clamped end (1). Ring valve geometries are generally so complex that the stress analyst has little idea of the location of the point(s) of peak flexural stress. Hence, there was an immediate need for a stress analysis procedure capable of locating these point(s) of maximum stress. The photoelastic coating method was selected as an applicable method for stress analysis of ring valves of this type.

APPLICATION OF PHOTOELASTIC COATINGS

Two methods of applying photoelastic plastic to the valves were investigated. The first method involved casting of liquid plastic directly onto the specimen in some type of mold. The second method entailed cementing a pre-cast plastic sheet cut to the desired outline directly to the specimen. The pre-cast sheet method was found to be the more satisfactory method for the following reasons:

1. Casting the liquid plastic was more time-consuming and required elaborate molds.
2. It was quite difficult to remove a valve coated with liquid plastic from the mold without inducing a separation between the plastic and the valve.
3. At the time the research was being conducted, circa 1966, commercially available sheet plastic had a 50% higher strain-optical coefficient than did commercially available liquid plastic.

This third reason is perhaps the most significant, since increased thickness is required for more sensitivity with plastics of equal strain optical coefficient. This is important, since the effect of the plastic on the stresses which would have existed in an uncoated valve becomes more difficult to assess as plastic thickness increases.

In order to successfully apply photoelastic plastic sheets to specimens having
intricate contours, it was necessary to develop refined procedures which included additional features to those recommended by the suppliers of the plastic and cement. These procedures are summarized below; the instructions given are intended to supplement, and not to replace, those furnished by the suppliers.

1. Materials. The plastic and cement listed are those found most satisfactory at the time this experimental work was conducted. PS-1-D plastic sheets (Photolastic, Inc., Malvern, Penn.), 0.010 inches or 0.020 inches thick. 4HRCCT Cement (Instruments Division, The Budd Company) or High Elongation Cement (Photolastic, Inc.).

2. An outline of the shape to be coated is traced onto the protective coating on the plastic. This pattern is cut out and trimmed such that it is slightly larger than the specimen. (Leaving the plastic oversized is necessary to ensure that the plastic covers the entire specimen after final trimming.) The protective coating is removed and the prepared plastic is cleaned with alcohol and set aside.

3. The surface of the specimen is cleaned using successive applications of Metal Conditioner and Neutralizer (Wm. T. Bean, Inc.) in much the same manner as surfaces are prepared for strain gage application. After each application of a chemical, the surface is purged with water before the remaining chemical is wiped away with tissue paper. The cleaned specimen is set aside.

4. The cement is prepared according to the suppliers instructions.

5. The cement is carefully applied to the specimen such that all areas are covered. The specimen is placed on a surface to which the cement will not adhere and the plastic placed atop the cement.

6. The plastic is positioned such that once final trimming to size is accomplished, all areas of the specimen are covered. Excess cement is squeezed out by pressing lightly on the plastic.

7. Final trimming such that the plastic and specimen boundaries coincide exactly is done using small files. Filing should be done toward the specimen to preclude separations.

The coated specimen is now ready for testing. Using the procedure outlined above, it was possible to coat specimens leaving negligible residual stresses and to test them with no separation at either the plastic-cement or cement-steel interfaces. A typical fringe pattern is shown in Figure 2.

THEORETICAL CORRECTION FACTORS

The need for a thorough study of the effect of the plastic became apparent during early attempts to measure stresses in ring valves. When 10- and 20-mil plastic coatings were applied to identical valves and the indicated stress corrected using the most applicable theory (reviewed below), the stresses indicated by the 10-mil plastic were as much as 40% higher than those indicated by the 20-mil plastic.

Review of the Literature. When the photo-stress technique is used to measure bending stresses in valves, the indicated stress on the valve surface as found from the photo-stress technique is not the stress which would have existed on the surface of an uncoated valve under identical loading conditions. Three phenomena account for the difference [2]:

1. The neutral surface in the valve is shifted toward the side on which the plastic is applied.

2. A strain gradient (which may, in fact, not be linear) exists in the plastic, and the measuring equipment indicates only an average strain through the plastic thickness.

3. The plastic bears a portion of the applied load and thus exerts a reinforcing effect.

It is the third phenomenon which has lead to the term "reinforcing correction factor". In the literature this term has been used to describe the effects of all three phenomena. A reinforcing correction factor relates the indicated strains in coated specimen to the strains in the same specimen identically loaded but without the plastic coating.

Many articles discussing reinforcing correction factors have appeared in the literature in the last several years. In the case of plane and/or flexural stresses, Zandman et al [2] have developed expressions based on a strength of materials approach. Other authors such as Duffy [3], Lee et al [4], Theocaris and Dafermos [5], Duffy and Mylonas [6] and Amba-Rao [7] have pointed out that a more general theory of elasticity approach is required to accurately assess the effect of the plastic in the presence of high strain gradients.
surface curvature, and/or ratio of plastic thickness to specimen thickness.

Since the valves analyzed were quite thin (0.008 inches) and since a minimum plastic thickness of 0.010 inches yielded less than three fringes, the plastic/specimen thickness ratio was necessarily appreciable. Also, since the valve geometries were intricate, high strain gradients were encountered. Thus, there seemed to be reason to question the applicability of reinforcing correction factors based on a strength of materials approach, and a thorough study of available theory was undertaken.

The more general theory of elasticity approach is presented for two special cases by Duffy [3] and in a third special case by Lee et al [4]. Duffy considers two types of one-dimensional sinusoidal displacements at the coating-specimen interface and derives correction factors for each assumed displacement; these correction factors enable one to evaluate the indicated strain if the deflections are known and are of the form(s) considered. The inverse problem of finding the deflections (from which, of course, strains follow immediately) given the indicated strains, is the problem of concern in this application, and this was covered only briefly at the end of Duffy's paper. Unfortunately, the results presented had primarily academic value for quantitative analysis of the valve geometries and loading conditions under consideration. The reference did, however, serve a valuable role: to indicate that care must be taken in interpreting indicated strain readings when strain gradients are high, and specimen thickness is small relative to plastic thickness.

The paper by Lee et al [4] is very similar in nature to Duffy's paper in that infinite coatings in which only the coating thickness is limited are discussed. Duffy's solution is extended to a two-dimensional deformation having radial symmetry. Once again, the applicability of the results depends upon knowing the displacements.

Thocaris and Dafermos [5] present an analysis for finite coatings for several specialized displacement distributions; e.g., one of them is a triangular distribution about an axis of symmetry. Modification of Zandman Theory. Because of the rather specialized nature of the geometries considered in most of these papers, it was decided that the most applicable theory was that presented by Zandman et al [2] which considers the problem of relating strains in an uncoated place in flexure to the strains indicated in a coated plate. Expressions are developed giving the ratio of indicated strain in a coated plate to strain in an uncoated plate subject to either the same load or to the same deflection. This ratio is plotted versus the ratio of plastic thickness to plate thickness, neglecting cement thickness. Because of the presence in the operating compressor of a valve stop to limit valve travel, the equal deflection condition is of interest here.

In this research cement thickness is, of necessity, of the same order of magnitude as the plastic and valve thickness, and consequently it was necessary to modify the results of Zandman's work to include the effects of cement thickness. A derivation based on Zandman's approach, but modified to account for the effect of cement thickness, follows. The notation is based on that used by Zandman.

Consider the element of a plate in flexure as shown in Figure 3 where \( t_s, t_a, \) and \( t_p \) are specimen, cement and plastic thicknesses respectively, \( \delta \) is the location of the neutral surface below the cement-specimen interface, and \( \epsilon_x, \epsilon_y \) are strains in the \( x- \) and \( y- \) directions, respectively. Under the assumptions that a straight line normal to the neutral surface remains straight after flexure, and that strains are inversely proportional to the radii of curvature, we may write \( \epsilon_x = z/R_x, \epsilon_y = z/R_y \) where the \( R \)'s are radii of curvature.

The neutral surface location is obtained from equilibrium considerations. Here it will be assumed that the same modulus of elasticity, \( E \), and the same Poisson's ratio, \( \mu \), exist for the cement and the plastic [8]. From equilibrium in the \( x \)-direction

\[
\sigma_x(s)dz + \sigma_x(c)dz = 0 \quad (1)
\]

where \( \sigma_x(s) \) and \( \sigma_x(c) \) are stresses in the \( x \)-direction in the specimen and coating (cement and plastic). Using Hooke's Law, Equation (1) becomes

\[
\int_{\delta_o}^{\delta} \left[ E \left( \mu \right) \right] (s)dz + \int_{\delta_o}^{\delta} \left[ E \left( \mu \right) \right] (c)dz = 0 \quad (2)
\]
where \( \delta' = t_s - \delta \) and \( \delta'' = \delta + t_a + t_p \).

Assuming \( \mu_c = \mu_s \) and integrating yields

\[
\delta = \frac{(t_s^2 - \lambda(t_a + t_p)^2)}{2[t_s + \alpha(t_a + t_p)]}
\]

(3)

where

\[ \alpha = E_c/E_s \]

The strain difference is

\[
\varepsilon_x - \varepsilon_y = \frac{1}{R_x} - \frac{1}{R_y} \]

(4)

The surface strain on the uncoated plate is at \( z = t_s/2 \). The average strain difference through the plastic is that which is indicated by the reflection polariscope. Under the assumptions of this derivation, this is the strain difference at the midpoint of the plastic where \( z = \delta + t_a + t_p/2 \). Let \( \varepsilon_x(0) - \varepsilon_y(0) \) be the strain difference at the surface of the uncoated plate and \( \varepsilon_x(c) - \varepsilon_y(c) \) be the average strain difference in the photostress plastic on the coated plate. From Equation (7)

\[
\varepsilon_x(0) - \varepsilon_y(0) = \frac{t_a}{2} \left( \frac{1}{R_x} - \frac{1}{R_y} \right)
\]

\[
\varepsilon_x(c) - \varepsilon_y(c) = \frac{1}{R_x} - \frac{1}{R_y}
\]

Now the assumption will be made that the radii of curvature are the same for equal deflections of coated and uncoated plates. Defining \( C_{zed} \) as "indicated average strain difference in coated plate" divided by "strain difference at surface of uncoated plate" yields

\[
C_{zed} = \frac{(\delta + t_a + t_p/2)/(t_s/2)}{(\delta + t_a + t_p/2)/(t_s/2)}
\]

(5)

where \( \delta \) is given by Equation (3) above.

Figure 4 shows \( C_{zed} \) plotted as a function of \( t_a/t_s \) and \( t_p/t_s \) for \( \alpha = 0.012 \), a value appropriate for steel and the plastic coatings used in the experimental portion of this work.

Equations (3) and (5) may be used to calculate a reinforcing correction factor which accounts for the effect of the cement. Note that two phenomena contribute to this factor:

1. The neutral axis is shifted by the plastic.
2. The polariscope indicates the average strain difference through the plastic.

For assumed equal deflections of the coated and uncoated plates, the reinforcing effect of the cement and plastic does not influence the indicated strain; thus the term "reinforcing correction factor" is not strictly accurate here. This term, however, has been used when the effects of the plastic on flexural stresses indicated by the photostress method of stress analysis are being discussed. The correction factor for equal loads is derived in Appendix A.

The equation corresponding to (5) which assumes a negligible cement thickness is

\[
C_{zed} = \frac{(\delta + t_c/2)/(t_c/2)}{(\delta + t_a + t_p/2)/(t_s/2)}
\]

(6)

where \( t_c \) is the coating thickness (plastic only), and

\[
\delta = \frac{(t_s^2 - \alpha t_c^2)}{2(t_s + \alpha t_c)}
\]

(7)

EXPERIMENTAL EVALUATION OF MODIFIED ZANDMAN THEORY

Although the modified Zandman theory is based on assumptions which more general theory has shown may not be completely realized for the valve geometries considered, it represented the most useful approach available for this application.

Because of doubts about the assumptions upon which the modified correction factors are based, an experiment to assess their accuracy was conducted. It was decided that it would be a relatively straightforward procedure to design a cantilever beam having the same thickness as the valves and having a discontinuity which would serve as a stress raiser and yield strain gradients similar to those in the valves. Comparison of Section A-A of the valve geometry being analyzed shown in Figure 5 to Section B-B of the cantilever strip shown in Figure 6 will reveal the reason for the choice of this particular cantilever valve geometry. (At this point several valves had been coated successfully and the locations of the points of peak stress as shown in Figure 5 were known.) Plastic of the same thickness(es) used on the valves would be applied to the cantilever strip and, seemingly, the reinforcing effect for the cantilever strip would be very nearly the same as for the valves.

At this point an interesting sidelight developed. A discrepancy of about 50% was found to exist between Peterson's [9] and Lipson's [10] results for stress concentration factors for thin plates in transverse flexure having semi-circular notches. For this reason the second calibration strip geometry shown in Figure 7 was chosen. Again a discrepancy existed, but the conflict was capable of resolution for the case of circular holes. Examination of
the pertinent literature [11, 12, 13] on which the stress concentration factors for circular holes should be based, revealed that Peterson's results were the ones which were well-founded. Thus, we used Peterson's factors as the basis for the experimental study of reinforcement correction factors and we recommend that Peterson's be the ones used rather than Lipson's for the cases studied here.

Beams of each geometry having thicknesses of 0.008 inches (the valve thickness) and 0.0125 inches were fabricated. Since the correction factor is a function of the plastic/specimen thickness ratio, it was decided to coat the 0.008 inch thick beam with 20-mil plastic and the 0.0125 inch thick beam with 10-mil plastic to obtain a wide range of thickness ratios. Care was taken to ensure that accurate values for the various thicknesses were known. The beams were loaded, given known deflections and the fringe order at the edge of the discontinuity plotted versus beam deflection. The linearity of the fringe order-deflection curves and the fact that they passed through the origin gave confidence that the plastic was applied with a negligible residual stress level. Figure 8 shows one of the curves obtained.

A theoretical stress \( \sigma_B \) which would exist at the edge of the discontinuity in an uncoated beam was calculated using the beam flexure formula and a stress concentration factor from Peterson's book [9]. An experimental stress \( \sigma_I \) indicated by the photo-stress technique was computed from the fringe order and plastic thicknesses. Note that \( \sigma_I \) is the uncorrected stress which is obtained when photostress plastic is applied to valves. An experimental reinforcing correction factor \( C_R \) was then computed from

\[
C_R = \frac{\sigma_I}{\sigma_B}
\]

This definition is consistent with the definition of the reinforcing correction factor given in the previous section as the ratio of the indicated principal strain difference in a coated plate to the principal strain difference in an uncoated plate, since the stress field at an edge is a uniaxial one for which \( \sigma = E \varepsilon/\left(1+\nu\right) \) where \( \varepsilon \) is the principal strain difference.

A reinforcing correction factor \( C_Z \) based on modified Zandman theory was calculated for each beam using Equations (3) and (5). The ratio \( R = C_E/C_Z \) was then computed, and the results summarized as shown in Table 1 for various free lengths of the cantilever beam; a standard deflection of 0.5" was chosen to compute the fringe order from the best straight line through the fringe order versus deflection data.

One aspect of Table 1 is particularly striking. Examination of the last column reveals that \( R = C_E/C_Z \) is generally between 1.35 and 1.50 for all cases considered. The average value of \( R \) for the higher range of correction factors seems to be only slightly higher than the average value for the lower range. No consistent trend in \( R \) as a function of the stress level or thickness ratio was found on the basis of the configurations established. It was established that the reinforcing correction factors based on the modified Zandman theory are definitely not sufficiently accurate for quantitative use. They may, however, indicate a stress far on the "safe" side of the actual stress; i.e., the stress in the uncoated valve computed using this theory can be considerably higher than the actual stress in an uncoated plate.

That the reinforcing correction factors based on a strength of materials approach are not quantitatively accurate is not surprising and is undoubtedly due to the fact that one or more of the assumptions upon which the derivation is based are not satisfied. Work by Amba-Rao [13] and Duffy [11] has shown that the assumption that normal strains in the coating vary linearly with distance from the neutral surface is most probably not valid for the valve geometries being considered, and thus the averaging effect is not correctly assessed. Zandman, in fact, states that "The generality of this result depends sensibly upon this condition".

Stress measurements were made on three valve geometries: the small radius and large radius valves shown in Figure 1, and a third valve having the inner radii even more enlarged. Both 10- and 20-mil plastic sheets were used to coat each valve geometry. Valve, cement, and plastic thicknesses were known to the nearest 1/10 mil for each of the four lobes of each valve. Each valve was deflected until it touched the valve stop and the indicated stress measured in each lobe of the edge of the valve. A separate correction factor based on the modified theory was used to correct the indicated stress for each lobe. Finally, all stresses were additionally adjusted by dividing by \( R = 1.42 \), an experimental correction factor based on the study using calibration strips. The four values of corrected stress for a given valve geometry coated with a given plastic were all within 10% of each other and generally within 5%. The averaged results of this testing program are summarized in Table 2. In the table, all stresses have been normalized by dividing...
them by the average stress in small radius valve coated with 10-mil plastic.

Further confidence in the coating techniques and the validity of the modified correction factor was obtained when strain gages were applied to four small radius and three large radius valves. Figure 9 shows a typical strain gage application on a small radius valve. The results of the strain gage study are summarized in Table 3 where the measured strains have been normalized by dividing by the average corrected strain in the small radius valve coated with 10-mil plastic. Note that the strain ratio agrees closely with those obtained with photoelastic plastic coatings. The strain magnitudes are slightly higher than obtained with the coatings; however, the strain gage readings were not corrected for the fact that the gages were not at the valve surface (which made the readings high) or for the fact that the gages indicate an average strain over their area (which made the readings low).

CONCLUSIONS

1. The coating procedure and materials presented above make it possible to coat thin flexural specimens having intricate contours, leaving no residual strains in the coating, and to repeatedly statically test the coated specimens with no separation at either the plastic-cement or cement-specimen interfaces.

2. Photoelastic coating analysis utilizing appropriate correction factors is a practical and accurate method of obtaining relative static stresses in thin flexural specimens.

3. The modified correction factor yielded corrected strains within 11-14% of each other for 10- and 20-mil coatings applied to the same valve geometry. Unmodified theory had yielded results differing by as much as 40%.

4. To get absolute strains, it was necessary to apply an additional, empirical correction factor. This experimental factor was obtained using specially prepared specimens having a geometry with a known stress concentration factor and as closely approximating the valve geometry as possible.

ACKNOWLEDGMENTS

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REFERENCES


APPENDIX

Correction Factor For Equal Applied Load Condition

Following procedures identical to those used by Zandman [2], one may obtain the correction factor for beams or plates subjected to equal loading conditions as

\[ C_{Zet} = \frac{\delta + t_a + t_p/2}{t_s} = C_{Zed} \gamma \]

where

\[ \gamma = \left[ 4(1+\alpha r^3) - \frac{3(1-cr^2)^2}{1+\alpha r} \right] \]

Here \( r = (t_a + t_p) / t_s \).

Figure 10 shows \( C_{Zet} \) as a function of \( t_a / t_s \) and \( t_p / t_s \) for \( \alpha = 0.012 \). It can be seen from this figure that \( C_{Zet} \) is not very sensitive to variations in \( t_a / t_s \) at particular values of \( t_p / t_s \) between 1.75 and 2.0. The implication to the experimenter is obvious; if a plastic can be chosen such that \( t_p / t_s \) lies in this range, the effect of adhesive thickness on the correction factor is minimal.
Figure 4. Correction Factor for Equal Deflection Conditions

Location of Maximum Stress

Figure 5. A Valve Geometry for Photostress Tests

Figure 6. Initial Calibration Strip Geometry

Figure 7. Second Calibration Strip Geometry

Figure 8. Fringe Order Versus Deflection for 0.0125" Beam with a Circular Hole

Figure 9. Fringe Order Versus Deflection for 0.010" Plastic

Figure 10. Correction Factor for Equal Load Conditions
Table 1. RESULTS OF EXPERIMENTAL STUDY OF CORRECTION FACTORS

<table>
<thead>
<tr>
<th>Type of Discontinuity</th>
<th>Beam Thickness</th>
<th>Beam Length</th>
<th>$\sigma_B$</th>
<th>$\sigma_I$</th>
<th>$\frac{C_E}{C_Z}$</th>
<th>$R$</th>
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<tbody>
<tr>
<td>Notch</td>
<td>0.008</td>
<td>2.022</td>
<td>37,000</td>
<td>223,000</td>
<td>6.02</td>
<td>4.32</td>
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<td>&quot;</td>
<td>&quot;</td>
<td>2.275</td>
<td>25,900</td>
<td>156,000</td>
<td>6.28</td>
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<tr>
<td>&quot;</td>
<td>&quot;</td>
<td>2.346</td>
<td>23,600</td>
<td>156,000</td>
<td>6.62</td>
<td>&quot;</td>
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<tr>
<td>0.0125</td>
<td>&quot;</td>
<td>2.165</td>
<td>71,400</td>
<td>224,000</td>
<td>3.14</td>
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<td>&quot;</td>
<td>&quot;</td>
<td>2.360</td>
<td>57,800</td>
<td>156,000</td>
<td>2.74</td>
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<tr>
<td>Hole</td>
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<td>42,500</td>
<td>291,000</td>
<td>6.85</td>
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<td>36,900</td>
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<td>31,700</td>
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<td>0.0125</td>
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<td>324,000</td>
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<td>94,700</td>
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<td>206,000</td>
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<td>58,200</td>
<td>192,000</td>
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Table 2. RESULTS OF PHOTOSTRESS TESTS

<table>
<thead>
<tr>
<th>Valve and Plastic</th>
<th>Maximum Stress (Corrected and Normalized)</th>
<th>Stress Ratio for a Given Plastic Thickness</th>
<th>Stress Ratio for a Given Valve Geometry</th>
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</thead>
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<tr>
<td>Small Radius Valve</td>
<td>1.00</td>
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<td>1.000</td>
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<tr>
<td>10-mil plastic</td>
<td></td>
<td></td>
<td>0.878 = 1.14</td>
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<tr>
<td>Small Radius Valve</td>
<td>0.878</td>
<td></td>
<td></td>
</tr>
<tr>
<td>20-mil plastic</td>
<td></td>
<td></td>
<td>0.808 = 1.11</td>
</tr>
<tr>
<td>Large Radius Valve</td>
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<td></td>
</tr>
<tr>
<td>10-mil plastic</td>
<td></td>
<td></td>
<td>0.987 = 1.11</td>
</tr>
<tr>
<td>Large Radius Valve</td>
<td>0.808</td>
<td></td>
<td></td>
</tr>
<tr>
<td>20-mil plastic</td>
<td></td>
<td></td>
<td>0.808 = 1.11</td>
</tr>
<tr>
<td>Very Large Radius Valve</td>
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<td></td>
<td></td>
</tr>
<tr>
<td>10-mil plastic</td>
<td></td>
<td></td>
<td>0.878 = 1.11</td>
</tr>
<tr>
<td>Very Large Radius Valve</td>
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<tr>
<td>20-mil plastic</td>
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<td>0.729 = 1.11</td>
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</table>

Table 3. RESULTS OF STRAIN GAGE TEST

<table>
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<th>Strain in Small Radius Valve</th>
<th>Strain in Large Radius Valve</th>
<th>Strain Ratio</th>
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</thead>
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<tr>
<td>1.06</td>
<td>0.92</td>
<td>1.16</td>
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Figure 9. Strain Gage Installation