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ADVISORY COMMITTEE ON REACTOR SAFEGUARDS
UNITED STATES ATOMIC ENERGY COMMISSION
WASHINGTON, D.C. 20545

February 21, 1968

MEMORANDUM

To : ACRS Members

H.E.

From : Harold Etherington, ACRS Member

Subject: OYSTER CREEK - PRESSURE VESSEL STUB TUBE FAILURES

This memorandum is prompted by the consideration that repairs to the Oyster Creek vessel will be essentially complete before the Committee has had an opportunity to study and appraise the stress analysis by the contractor and his subcontractors.

Reason for Concern

The basis for concern is that the contractor does not appear to be concerned that the stresses are extremely high.

Contractor's Statement on Cause of Failure. The contractor (Reference 1) attributes the problem to "chemical activation of the surface of sensitized stainless-steel stub-tube material contained generally within high-stress areas of the stub tube, the presence of defect-containing field welds between the stub tubes and control rod housings, and minor defects contained within the welds between several of the in-core instrument tubes and the vessel". There can be no quarrel with this conclusion as a statement of probable fact, but the statement fails to indicate that the stress is extremely and perhaps unacceptable high.

UNDERSTATED

Contractor's Statement of Cause of Stress. Reference 2 (Amendment 29) not only touches on stress quite lightly, but it attributes the damaging stress to the shop weld. On page 18, the Amendment refers to "All of the cracking indications associated with the shop welding ..."— this essentially means all of the cracks at the bottom of the stub tubes. I believe that, although the stresses from the shop operations are high, they are overshadowed by stresses from field welding. It was for this reason that measurements of tube diameters above and below the weld were requested and have been supplied by the contractor.

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However, Reference 1 (the February letter) states on p.3 that "experience and measurements of the distortion indicate that the field weld between the stub tube and the control rod drive housing applied fabrication induced strains which exceed the elastic limit of the material on the 'free' length of the 'hill side' on the outer row of the stub tubes". (The underscoring is mine.) This indicates a shift in the contractor's position, but he has not recognized (or has not acknowledged) that the calculated stresses on the long side, although smaller than those on the short side, also exceed the "elastic limit" by a wide margin. The proposed removal of weld metal (pp. 3 and 4) will have a beneficial effect on stress, partly (as stated) because it will increase the effective length of the stub tube, but also (not stated) because it will weaken the weld area sufficiently to permit yielding and relief of bending moment — this may be a dubious remedy.

The contractor now proposes (Reference 1) to remove the field welds. This is fine, but there will still be high residual stresses although of opposite sign. It is hoped that, in developing procedures for rewelding (page 40 of Reference 2), the laboratory will pay attention to avoidance of the excessive deformation evident from field measurements -- this is not mentioned in the reference.

It is concluded by GE, CE and Teledyne (Reference 1, p.3) that "the stresses tend to be compressive in nature in the hot operating condition". It is true that there would be substantial relief of stress from the shop weld at operating pressure, but again the statement appears to ignore the effect of the field weld.

Conditions for Stress Corrosion

This is a problem for the metallurgists. However, it is noted that on p.3 of Reference 2, it is found that 10 ppm of chlorides produced cracking of sensitized Type 304 stainless steel "stressed above yield" in 48 hrs. at 180° F. This raises a question: How much chloride would be required to have a similar effect in 350,000 hrs. at 550° F, and possibly at some higher stress and with other unpredictable variations? Will the proposed cladding of Type 308 L provide the desired protection and immunity?

Interpretation of Calculated Stress

Design calculations are based on elastic theory. In cases such as those under consideration, the calculated stress is fictitiously high, because plastic deformation limits the stress to a value somewhere above the yield stress but usually much below the ultimate strength. All stresses given in this memorandum are calculated from elastic theory. Calculation of true stresses would require a very complex elastic-plastic analysis.

As a more realistic first approximation to the true stress, the calculated stress can be converted to strain by dividing by the modulus of elasticity, 28×10^6 psi; for example, a calculated stress of 350,000 psi corresponds to a strain of 0.0125 in./in. In a plain carbon steel, the actual stress would be limited, by plastic deformation, to approximately the yield point. In an austenitic steel the actual stress would be considerably greater than the yield stress, and could be determined from the stress-strain curve.

The stress-strain relation for an austenitic steel is non-linear even below the yield stress. If the 0.2 percent offset yield strength for AISI Type 304 stainless steel is 35,000 psi, then the total strain, elastic plus inelastic, at the yield stress is 0.00325 in./in. This means that a calculated stress of 91,000 psi would in fact (to a first approximation) correspond to an actual stress of only 35,000 psi. This steel, however, has no sharp yield point, and stresses such as those calculated in this memorandum will lead to actual stresses much above the yield stress.

It is established that stress alone is the stress criterion in stress corrosion, or is strain also a factor?

Design Criteria

Some clarification of the design criteria might be in order. Page 36 of Reference 2 (Amendment 29) states the vessel design "complies with Section 1 ... of the Code, with applicable Nuclear Case Interpretations, ... In addition, the specification requires a detail stress analysis comparable to that required by the 1963 Edition of Section III Nuclear Vessels but with the material allowable stresses as set forth in Section I. .... The General Electric specification requires a fatigue analysis ... "

Allowable Stress. What is the maximum allowable local stress or strain? Unless it is proposed to remove all ceiling on developed stress or strain, short of tensile failure, some criterion would appear appropriate. In which category under Section III are the bending stresses treated.

Code Interpretation. For obvious contractual reasons, the Code considers a vessel to end at the nozzles and does not consider the effect of field welds. In this case the stub tubes and control rod drive housings form part of the pressure boundary of the vessel and, if the housings had been attached in the shop, the detail would have been analyzed and found unsatisfactory in design or welding procedure — at least the final assembly would have been stress relieved. It might be asked whether the intent of the Code has been satisfied on whether a loophole has been used by which the contractor has found it convenient to complete the pressure boundary in the field and has not applied the Code — obviously there was no intention to evade the Code, but it does appear that an unsatisfactory condition was not recognized by the designer or by the writer of welding procedures.

Fatigue Analysis. What residual stresses will be used in the fatigue analysis?

Attachments

Attachments A, B, and C give the gist of the calculated stress patterns expected in the control rod drive housings and stub tubes. The calculations are based on elastic theory with simple models and on the field measurements of the Oyster Creek and Niagara Mohawk vessels. The studies do not purport to be a serious stress analysis of the conditions, but rather a random probing into the causes and distribution of stress, with the object of raising some questions which may need answers. The methods of calculation are indicated only broadly; and details are omitted to avoid further increase in the bulk of this already lengthy memorandum.

Attachment A - General Formula for Ring Loaded Tubes and Reference Dimensions. This attachment gives the general formulas for a tube under radially symmetrical loading. (A brief explanation of the basis of these formulas was supplied to one ACRS member at his request., The explanation could be written up if others are interested.)

The attachment also gives reference dimensions from page 24 of Reference 2, and the derived section constants.

Attachment B - Stresses Induced by Field Weld. This attachment correlates stress calculations with field measurements. Calculated curves of distortion conform in shape to the measured distortion profile. The peak calculated stress in the control rod drive housings is 380,000 psi.

The peak calculated stress for a long stub tube is 388,000 psi. For short stub tubes it is many times greater, but the simple calculational model is poor for such tubes. The maximum stress in the stub tubes, tensile at the outer surface, is approximately 1 in. from the field weld, or at the shop weld if the length is less than 1 in.

It is confirmed that the stress from the field weld is sufficient to cause corrosion cracking at the observed location, and that this is true irrespective of stress from the shop weld.

Attachment C - Stress in Stub Tube from Shop Treatment. This attachment shows that calculated stresses from the shop weld, although high, are smaller than those from the field weld. The stresses arise from differential contraction in cooling from the stress-annealing temperature.

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Conclusions

1. The control rod drive housing and stub-tube assemblies as designed and fabricated are in a state of exceptionally high stress.

2. The maximum tensile stress at the surface of the stub tubes is at the shop weld where cracking is observed.

3. The stress from the field weld is greater than that from the shop weld.

4. The calculated stresses are fictitiously high and high local stresses are common in structures in the neighborhood of welds that have not been stress relieved. But, in a location where in-service inspection will be extremely difficult and repair even more so, a very high standard of quality assurance is necessary and the pre-existing stress condition appears to be unsatisfactory.

5. Very high stresses exist in the control rod drive housing as well as in the stub tubes. A failure of a housing could lead to partial control rod ejection.

Attachment B - Stresses Induced by Field Welds

This attachment discusses implications of the recently measured distortion of control rod drive housings, CRDH,(i.e., the inner through tubes) and stub tubes.

Introduction

The purpose is to calculate stresses by elastic theory consistent with the measurements of deformation.

References. The following references were used:

1. Amendment 29, "Oyster Creek Nuclear Power Plant No. 1, Status Report on Reactor Vessel Repair Program", December 4, 1967.

2. "Measurements of Four Control Rod Drive Housings at Field Weld", GE memo W. A. Kruse to H. C. Matteson, January 25, 1968. (These measurements are for the Niagara Mohawk vessel).

3. Three-sheet report of Oyster Creek vessel measurements dated February 7, 1968.

(References 2 and 3 were supplied with copies of a memorandum dated January 17, 1968, R. L. Tedesco through R. S. Boyd).

The Measurements. The requested measurements were the I.D. of the CRDH, taken at 1/2 in. intervals, above and below the weld; and in two orthogonal directions, corresponding to maximum and minimum variations. Similar data were requested for the stub tubes if possible.

The large amount of data supplied by the contractor covers the requested range very well, but no set of measurements covers all the data for any particular combination of housing and stub tube. The analysis is therefore presented piecemeal, but probably nothing is lost by this treatment.

Condition of Assemblies. Nothing is known of the history or general condition of the assemblies. Most have probably been straightened after welding and many have been ground to remove cracks. It is assumed that the CRDT diameters were reasonably uniform before welding, but this may not be true of the stub tubes, which would be distorted by the shop weld to the vessel and subsequent stress relief.

Control Rod Drive Housings

The desired analysis requires that measurements be made of maximum and minimum distortion in order to permit assessment of the effects of both radially symmetrical forces and the flattening forces arising from asymmetry of stub-tube length. Only the four measurements of the Niagara Mohawk vessel gave adequate data and these are used in the following analysis.

Radially Symmetric Distortion. It can be shown that the effect of bending moment exerted by the weld is small in comparison with the effect of direct radial force. The radial contraction is calculated as a function of distance from the weld and the results are plotted in Figure 1. The figure shows that the measured average radial deflection profiles for the Niagara-Mohawk housings conform to the shape of the profile calculated by elastic theory (wave length 6.14 in.). The maximum radial contraction (0.020 in.) in the calculated curve was selected empirically to match the measured maximum for CRDH's 18-07, 02-35, and 30-07. The measurements for CRDH 38-11 show somewhat less distortion but the shape of the profile is the same. The symmetry of measured data above and below the weld confirm the absence of any large bending moment induced by the weld.

Figure 2 shows the peak stress, calculated by elastic theory for a modulus of elasticity of 28 x 10^6 psi, to be 380,000 psi. The figure shows the rapid decay of stress typical of this type of structure.

The radial force at the weld for a radial contraction of 0.02 in. is 95,400 lb. per in. of circumference, calculated at mean radius. Like the calculated stress, this force is fictitiously high.

Figure 3 shows the data for the twelve outer-circle assemblies reported for the Oyster Creek vessel. The data appear to show a greater scatter than those for Niagara Mohawk and a tendency to more abrupt decay with distance from the weld. Factors that could lead to the condition include possibly poorer quality control of field welding procedures, an out-of-round condition (measurements were made on a single diameter), or possibly inferior quality of the data.

Out-of-round Distortion. The Niagara-Mohawk data provide measurements in the XX and YY directions for the four assemblies. The maximum ovality is as follows:

CRDH identification number	07-30	02-35	18-07	38-11
Maximum difference in diameter, in.	0.032	0.013	0.006	0.005

The directions of measurement in no case correspond to a radial-tangential orientation or otherwise suggest that they represent maximum and minimum diameters, but there does appear to be a tendency to elongation in the radial direction of the vessel, and contraction in the tangential direction, as a result of the flattening.

Calculations (not attached) show that the force required to produce the measured deflection is an order of magnitude less than the total distributed radial force and that the stress in still smaller. The differences in diameter for housing 30-07, which showed the greatest out-of-round, are plotted in Figure 4.

Stub Tubes

Stub-tube data are available only for the Niagara-Mohawk vessel. These data include measurements of tubes well within the array of tubes as well as tubes near the periphery. The asymmetry of stub-tube length increases towards the periphery. The analysis is made for various assumed lengths of stub tube, but the tubes are assumed to be uniform in length. A better model would involve a great deal more work.

Bending. The continuous curve in Figure 5 shows the calculated radial contraction as a function of distance for a tube of infinite length, and part of a similar curve for a tube 1.95 in. long, both for a maximum radial displacement of 0.08 in. at the field weld. Displacement curves have also been calculated but not plotted for lengths of 2.92 in., 3.89 in., and 4.86 in. In all cases, as suggested by Figure 5, the curves lie close to the curve for infinite length, except that departure occurs as necessary to meet the condition of zero displacement at the terminal point corresponding to the fixed (shop weld) end of the stub tube. In summary, for a given displacement at the weld, the calculated deflection curve is only slightly dependent on length for stub tubes two inches long or more. For still shorter tubes more radical departure is necessary to meet the end condition.

As reported for the CRDH Niagara-Mohawk data, the stub tube measurements in Figure 5 show a wide scatter, but are not inconsistent with the calculated conclusions. Measurements for tubes well within the array are plotted with crosses, and, as would be predicted from the analysis, these fall within the pattern of data for tubes near the periphery.

The calculated radial inward force on the stub tube and the maximum bending stress are as follows:

Length of stub tube, in.	Radial force per inch of average circumference, lb/in.	Maximum bending stress, lb/in.^{2}
Infinite	91,100	388,000
5.84	91,100	388,000**
3.89	92,200	398,000
1.95	99,200	498,000*
0.974	316,000	2,870,000*

* Calculated at a distance 0.974 in. from weld, not necessarily maximum stress.

The table shows that the force and stress depart significantly from the values for infinite length only when the tube is less than four inches long, but that for lengths under two inches a radical increase in force and stress occurs.

The maximum bending stress, tensile on the outside, is approximately one inch from the end of the tube, or at the shop weld if the length is less than one inch.

Other Studies

The stresses in the field weld have been explored semi-quantitatively and also the effect of shear forces on stress and strain in very short tubes. Nothing was found that casts doubt on the credibility of the condition as described. If, however, part of the distortions were caused by hot plastic deformation, the stresses would be lower.

An attempt at analysis of a stub tube of unsymmetrical length was abandoned as being too cumbersome.

ATTACHMENT C - STRESSES IN STUB TUBES FROM SHOP TREATMENT

This study is less extensive than that for the field weld because it is understood that a detailed stress analysis has been made by the contractor and because the calculated stresses are smaller than those from the field weld.

Nature of Stress

The shop-induced stress is caused by differential contraction of the austenitic steel stub tube and the pearlitic steel vessel (and Inconel weld) during cooling from the stress annealing operation. Suppose the vessel and stub tube are essentially stress free at the stress relieving temperature, i.e., omit the refinement of allowing for relatively small residual stresses in the base materials and the larger stresses probable in the Inconel weld. If the stub tube and vessel were free to contract independently, the outside radius of the stub tube would be 0.010 in. smaller than the corresponding opening in the vessel. The stub tube is therefore pulled out as shown in Figure 1 a.

The behavior will be between two extremes:

1. The tube may be held rigidly as in Figure 1 b.
2. The weld may yield and relieve the bending moment as in Figure 1 c.

The actual behavior, for a stub tube of given dimensions, will depend on the strength of the weld and the length of stub tube projecting on each side. If the weld does not yield, or if several inches of tube project on each side, condition 1 b prevails. If the weld yields, which is more likely if the weld is shallow or if only a short length of tube protrudes on one side, then condition 1 c prevails.

Case 1. Stress in Stub Tube with Rigid Constraint. For the case shown in Figure 1 b, constants of the general formula of Attachment A are A = B = 0 and C = D = 0.01.

P = 22,800 lb per inch of circumference M = 14,100 in. lb. per inch of circumference.

Maximum stress = 150,000 psi (at the weld, and tensile on the outer surface). In addition there will be a hoop tensile stress of about 23,000 psi.

For a tube projecting at least a few inches on each side of the weld, the condition is the same except that P is doubled.

Stress in Stub Tube with Free End. For the case shown in Figure 1 c, constants of the general formula are A = B = D = 0 and C = 0.01.

P = 11,500 lb per inch of circumference. The maximum bending stress is 50,000 psi at 0.974 in. from the weld and is compressive at the exposed surface.

Intermediate Cases. Stresses for cases with only a short length of stub tube protruding from one side of the weld, and cases of asymmetric bending because of weld yielding are easily calculated, but will not approach the high stress levels caused by the field weld.

Weld Strength. The maximum bending stress in the weld is given by $S = 6\ M_o/d^2$, where d is the effective depth of the weld measured parallel to the stub tube axis. The direct stress is P/d, and the maximum tensile stress is the sum of the two components.

At the high side of the weld d appears to be about two inches. At the low side, allowing for poor support by the fillet, the effective depth may be about one inch.

For Case 1, likely to apply to the high side, the maximum tensile stress in the weld is 32,500 psi, which is well below the yield stress of Inconel (55,000 psi) and just below the yield stress of the stainless steel (35,000 psi). For Case 1 with a through tube, the tensile stress is 23,000 psi.

At the low side, Case 1 would lead to a maximum calculated tensile stress of 107,000 psi, and the stainless steel (and possibly the Inconel) would yield. The condition will therefore shift towards Case 2, with a direct tensile stress of 11,500 psi and a residual bending moment.

Direction of Intergranular Penetration. Case 1 of Attachment C approximates the high side of the shop weld with a short protruding end of the stub tube. The bending moment is sufficient to cause yielding in the outer surface of the tube near the weld, and may be sufficient to cause yielding in the stainless steel along the weld interface. This would explain division of the crack into two branches with one parallel to the weld interface.

The above description applies equally to the bending moment caused by the field weld, except that "may be" becomes "will be".

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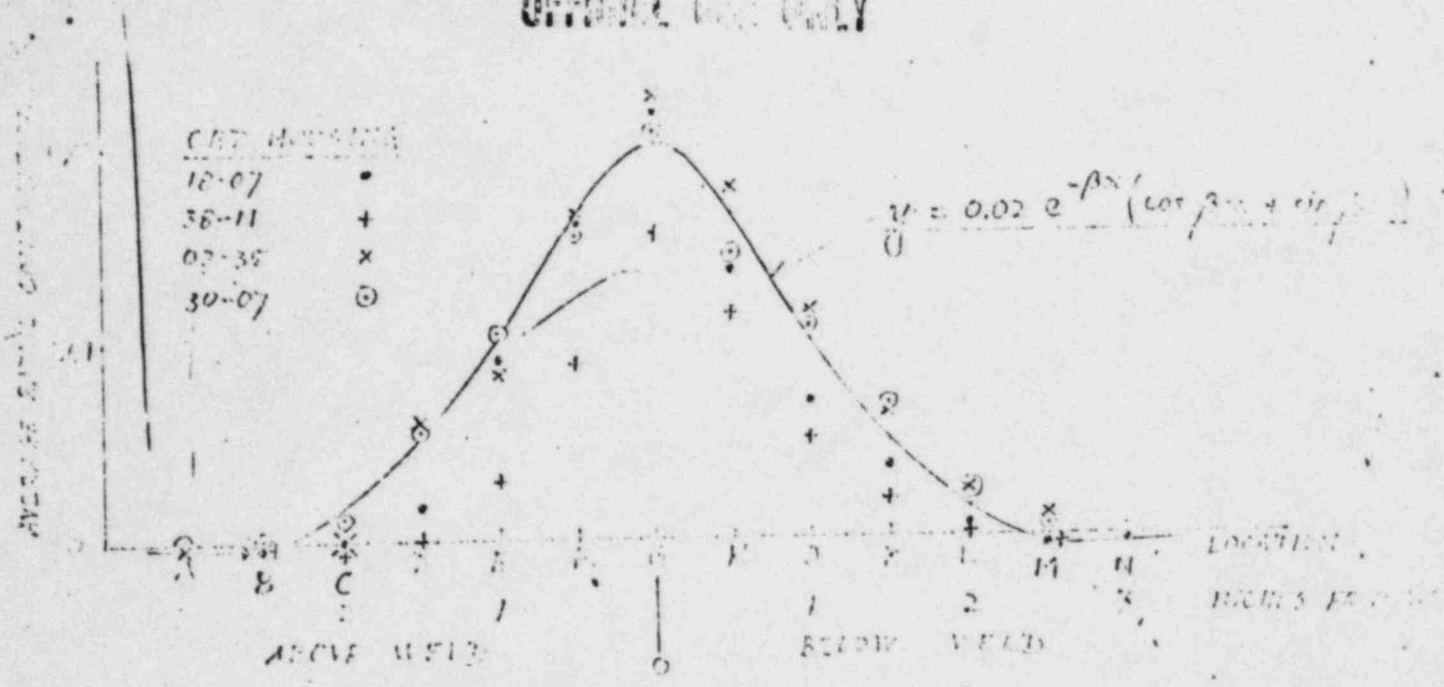

FIG 1 CRD HOUSING : RADIAL CONTRACTION

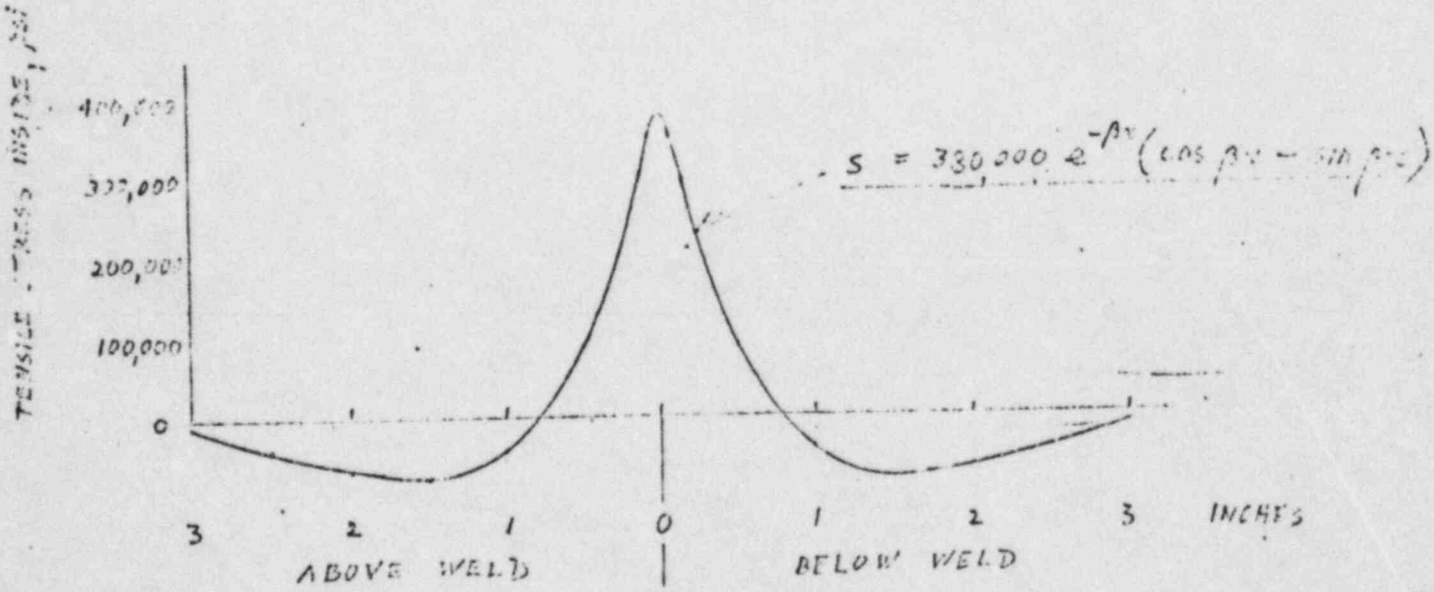

FIG 2 CRD HOUSING : CALCULATED STRESS

NIAGARA MOHAWK

(ATT. B)

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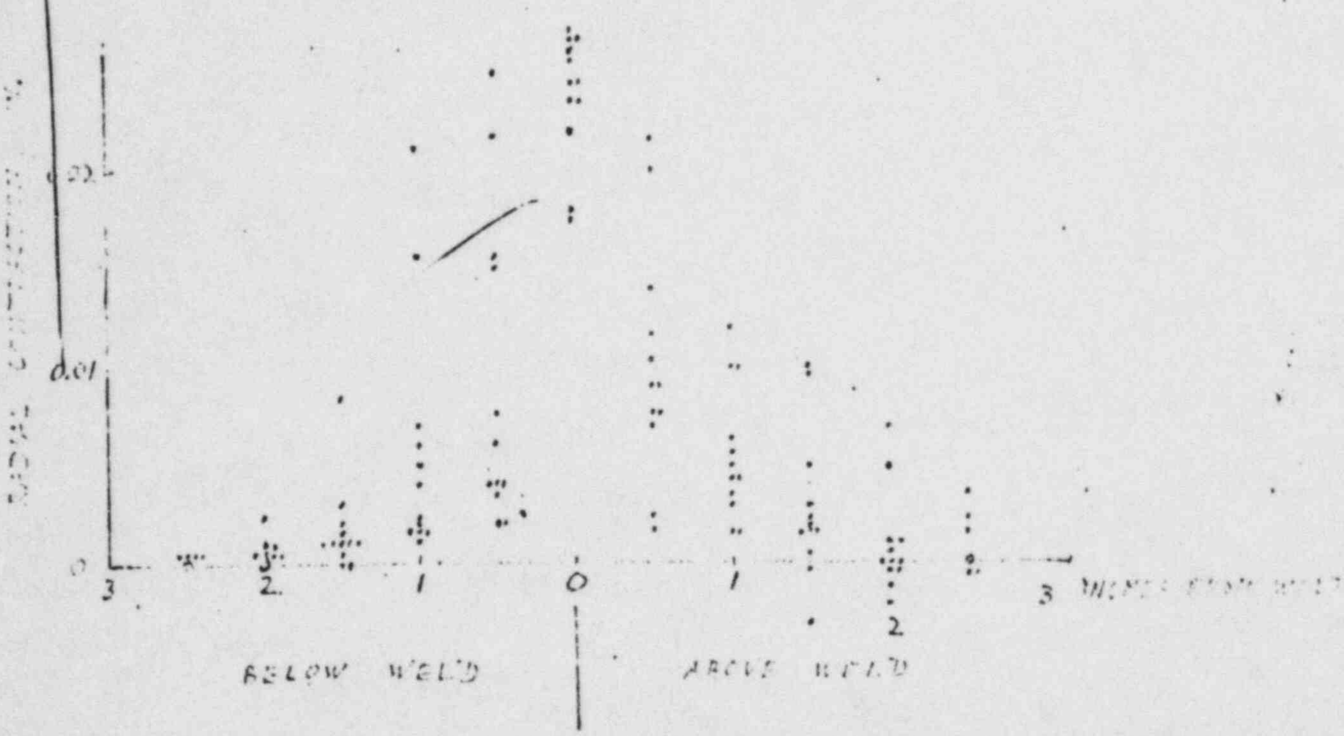

FIG 3 CRD HOUSING : RADIAL CONTRACTION

OYSTER CREEK

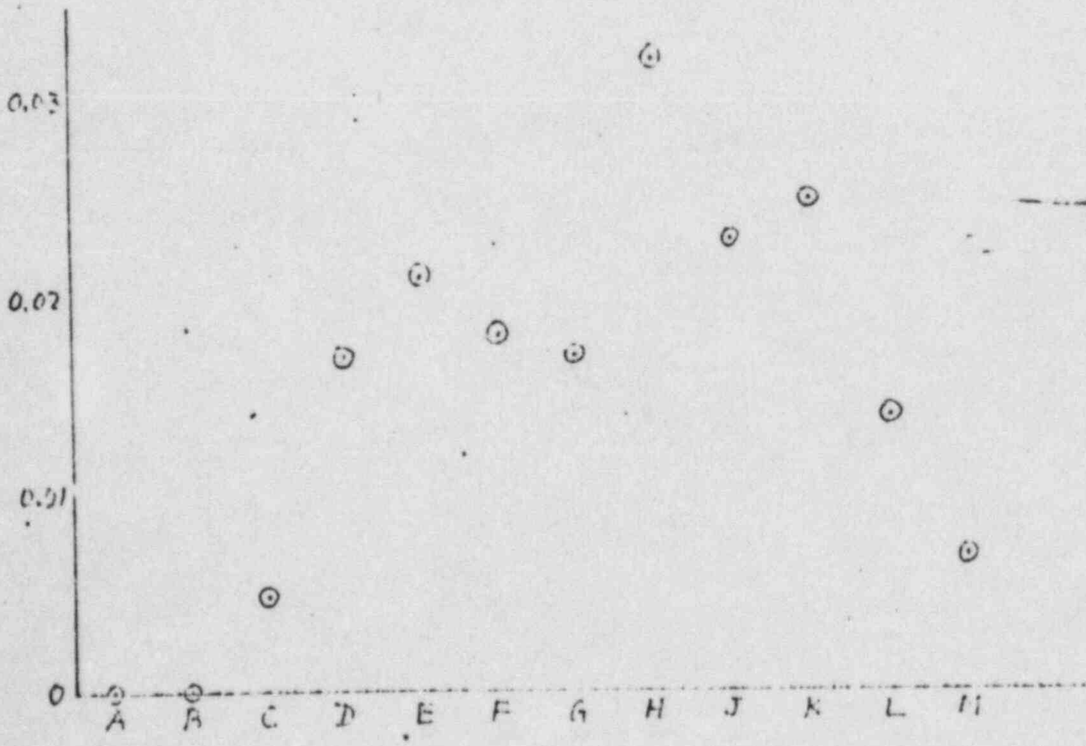

FIG 4 CRD HOUSING : (OUT-OF-ROUND)

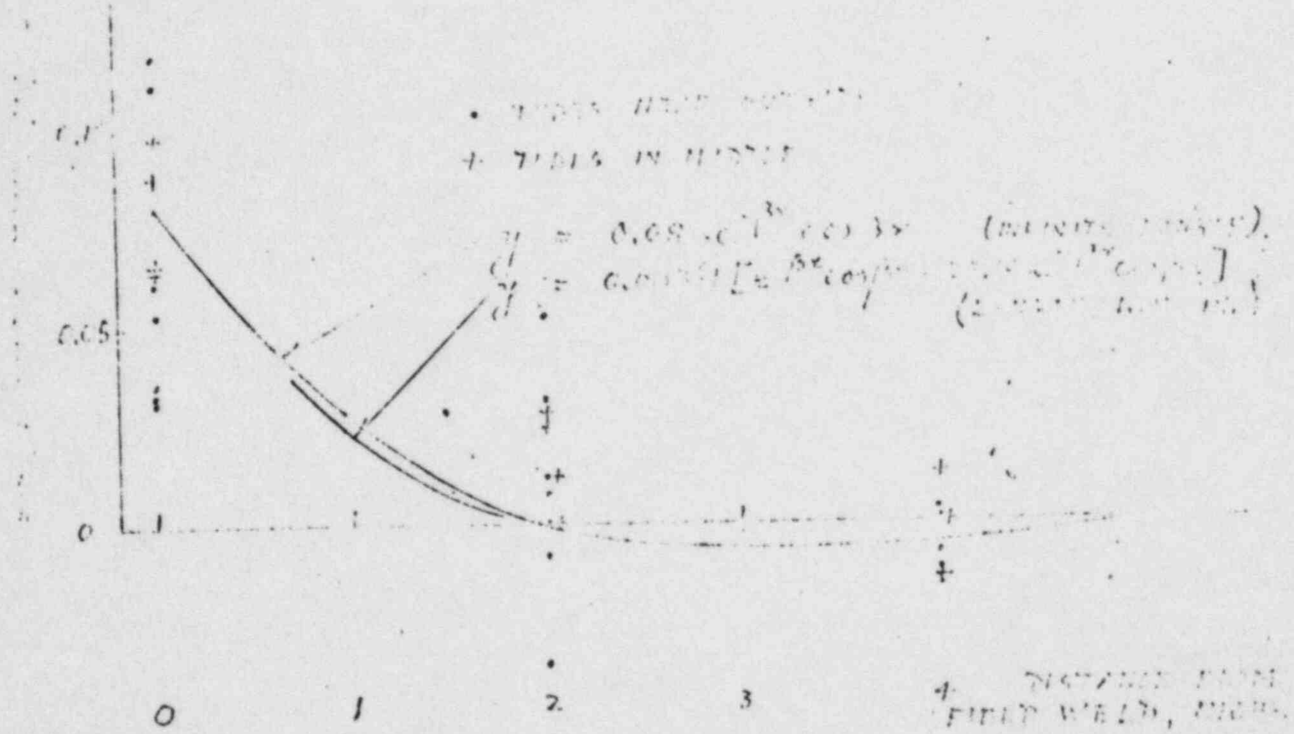

FIG. 5. STUD TUBE: RADIAL CONTRACTION

NIAGARA MOHAWK

(ATT. I)

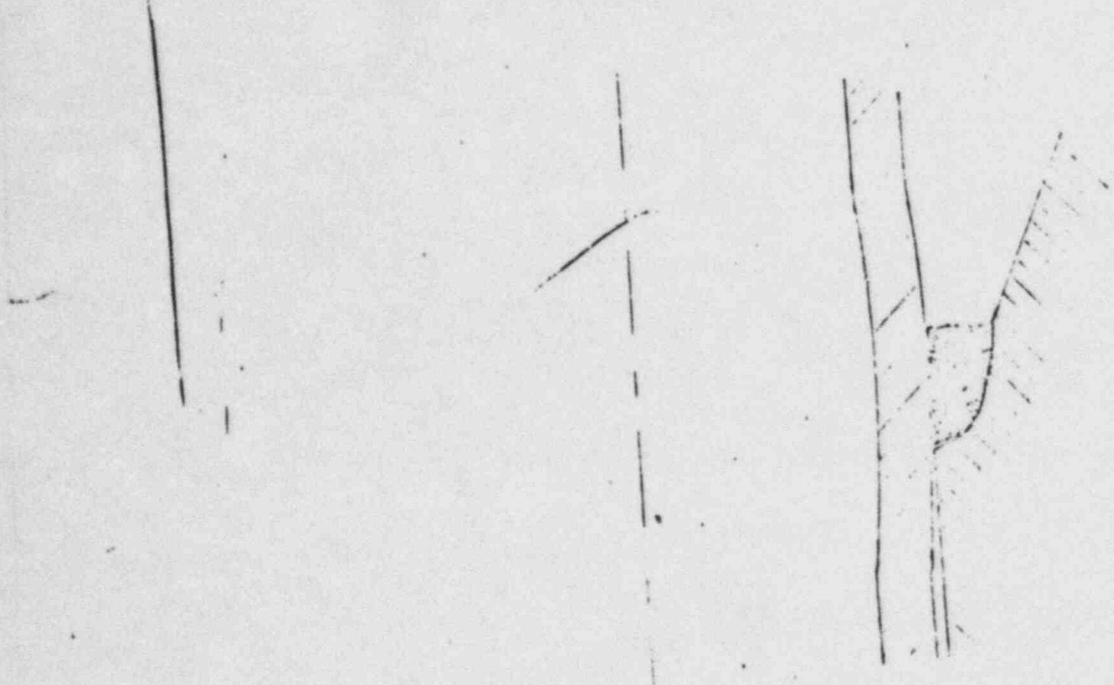

FIG 1a

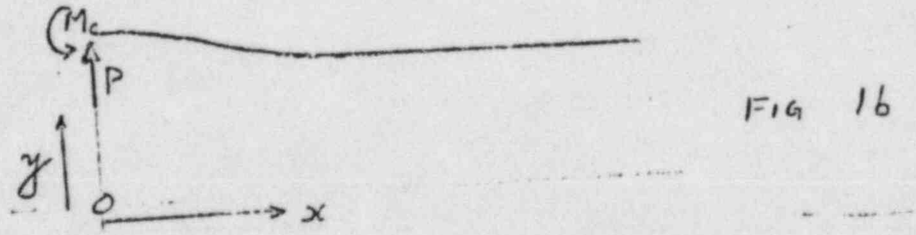

FIG 1b

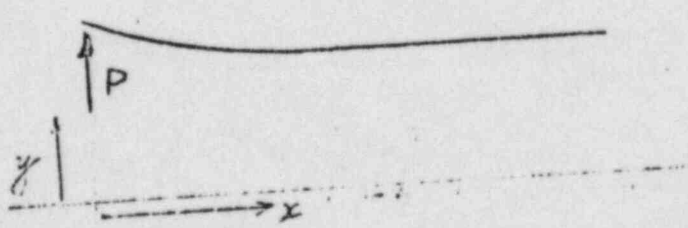

FIG 1c

FIG 1 ATTACHMENT C