STRUCTURAL EFFECTS

OF

ATWS OVERPRESSURE TRANSIENTS

IN

BYRON JACKSON PRIMARY PUMPS

BABCOCK & WILCOX CO.

NPGD

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CONTENTS

SUMMARY

The effect of an ATWS transient on the structural integrity of primary pumps manufactured by Byron Jackson for Babcock & Wilcox nuclear reactor is evaluated in this report. The primary effect of the transient on the pump structure is to cause a pressure surge that peaks at less than 3750 psi.

The effect of the transient is evaluated by examining stresses caused by a static pressure of 3750 psi. Also, a quantitive estimate of the deformation is made to determine the potential for interference with rotation of the impeller.

It was found that stress in the flanges, the closure, and those portions of the pump in which the bearings are mounted are within allowable stresses in accordance with ASME Nuclear Pressure Vessel Code for Level C incidents. Stress in portions of the volute and the diffuser vanes do exceed allowable stress for Level C incidents. The location and magnitude of these stresses are presented in this report. These stress levels are well below material rupture strength, and closure bolts are within allowable stress limits; therefore, it is concluded that the structural pressure boundary will not be violated.

An evaluation of deformation resulting from these stresses shows that displacements which could affect the rotating impeller are small compared to available clearances.

This provides assurance that the structural integrity of the primary pump will be retained, and the post-ATWS function of the primary pumps will not be impaired by an ATWS pressure pulse, which was determined in accordance with current NRC guidelines.

INTRODUCTION

SCOPE

This document addresses the effect of an ATWS pressure transient on the structural integrity of the primary reactor coolant pumps built for the reactors listed in Table 1.

TABLE 1

REACTORS USING BYRON JACKSON PUMPS

UTILITY	REACTOR
Consumers Power Co.	Midland 1 Midland 2
Arkansas Power & Light Co.	ANO-1
Florida Power Corp.	Crystal River 3
Toledo Edison Co.	Davis Besse 1

A description of the pump and its stress analysis is included. A comparison of stress in the pressure boundary caused by the worst ATWS transient is made with allowable stress for Level C incidents. A quantitative estimate of distortion is included to determine potential effects on impeller clearances. Flange seal capabilities are also evaluated.

Earlier evaluation of the pump is included in topical report BAW 10099. That report determined the pressure at which stress reached allowable values for emergency conditions. The re-evaluation herein is based on current revised guidelines from NRC for ATWS evaluation.

ASME Allowable Stress

Allowable stress for Level C service limits are given in Subsection NB of Section III, Division 1 of the ASME Pressure Vessel codes. The

limiting requirements for this application requires that primary membrane stress not exceed 1.2 S_m, and that primary membrane plus primary bending stress not exceed 1.8 S_m.

The pump is made from austenitic stainless steel casting per the requirements of ASTM A-351 Grade CF8M. The value of S_m for this material at 680°F is 16,400 psi. Therefore:

Primary membrane allowable stress = 19700 psi

Primary membrane + primary bending allowable stress = 29500 psi

Irradiation Effects

The primary pumps are sufficiently distant from the reactor to make irradiation effects negligible, therefore material properties of structural components in the pump do not include any modification for irradiation effects.

ATWS Transient

The pressure transient resulting from an ATWS event has been determined using the NRC guidelines of Alternate Number 3 specified in Volume 3 of NUREG 0460. The results of this study are included in a letter report bearing the title "Analysis of B&W NSS Response to ATWS Events", January 1980.

The above report shows the pressure transient peaks at a pressure less than 3750 psi, and that the coolant temperature does not exceed 660°F. The pump stresses are evaluated at 3750 psi, and the allowable stress for the pump materials is based on a temperature of 680°F.

PUMP DESCRIPTION

The Byron Jackson pumps are vertical, single stage, single bottom suction, horizontal discharge, centrifugal diffuser casing units with controlled leakage mechanical seals. They are driven by squirrel cage, vertical

AC induction motors. Some of the design features of this pump are listed below.

TABLE 2 - PUMP DESIGN PARAMETERS

Design Pressure	2,500 psig
Design Temperature	$650^{\circ}F$
Operating Pressure	2,250 psig
Rated Flow - 4 Pumps Operating	88,000 gpm
Maximum Flow	130,000 gpm
Developed Head - 4 Pumps Operating	327 feet
Suction Nozzle Inner Diameter	28 inches
Discharge Nozzle Inner Diameter	28 inches
Power Source	6,600 volts, 3 Phase, 60 Hz
Motor Horsepower at Operating Temp.	6,090 HP
Rotating Speed	1,185 rpm

The pump assembly consists of the motor, the driver mount, the rotating element (coupling, impeller, and shaft), the heat exchanger, the seal cartridge (and mechanical seals), the cover plate, and the case.

Each reactor contains four of these pump assemblies for the purpose of recirculating primary reactor coolant. The pumps are installed in the cold leg of the piping system between the steam generator and the reactor vessel.

The general arrangement of the pumps and sketches showing pertinent details of the pump are shown in Figures 1, 2 and 3.

GENERAL DESCRIPTION OF MATHEMATICAL MODEL

The pump is modeled in two parts: pump case and case closure. The pump case model is a three-dimensional finite element model. This model uses solid elements for the flanges and diffuser vanes. Shell elements are used for the volute, and for suction and discharge nozzles.

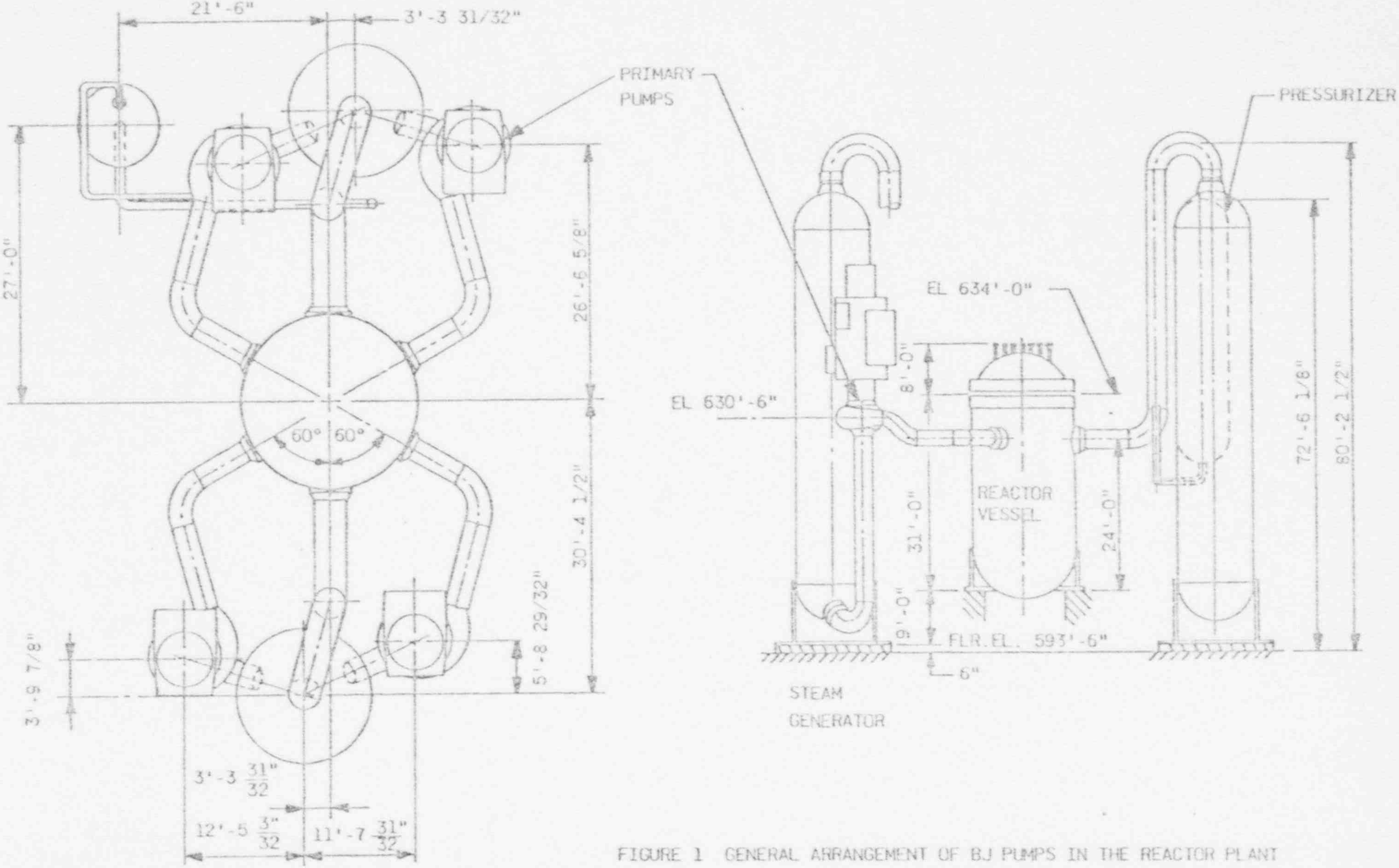


FIGURE 1 GENERAL ARRANGEMENT OF BJ PUMPS IN THE REACTOR PLANT

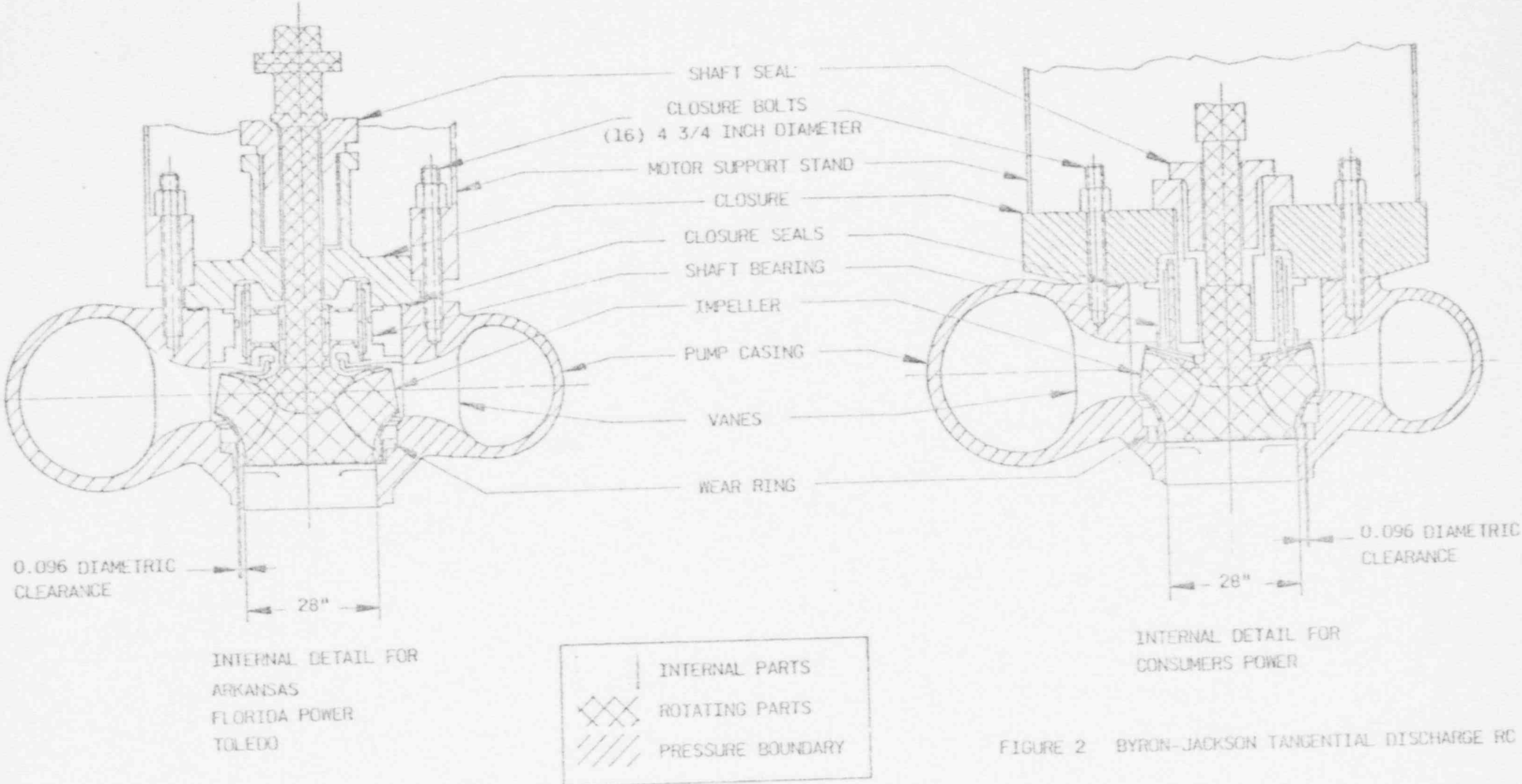


FIGURE 2 BYRON-JACKSON TANGENTIAL DISCHARGE RC PU

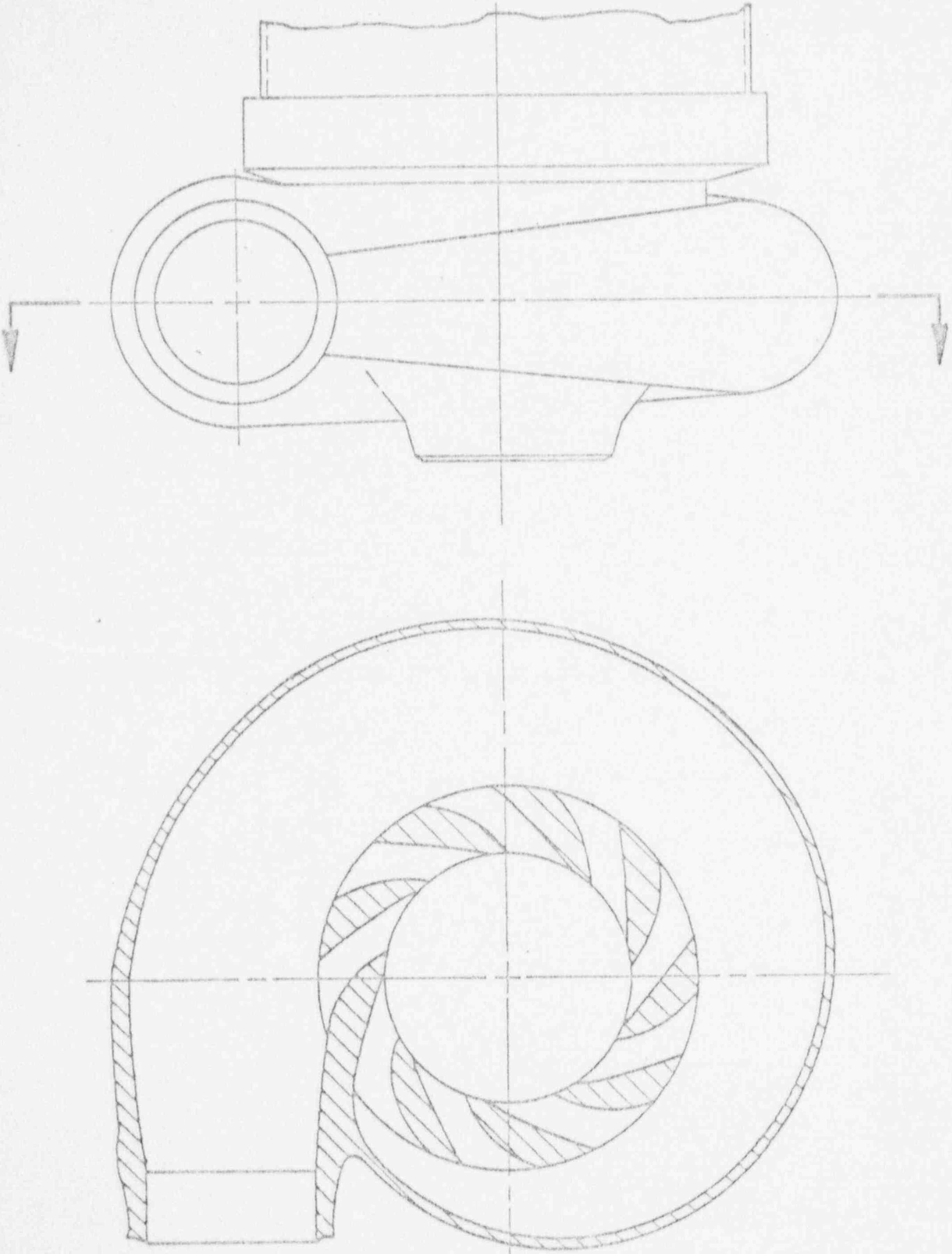


FIGURE 3 BYRON JACKSON PUMP CASING

The case closure is a symmetric structure. It is modeled using axisymmetric two-dimensional finite element techniques. Each model includes portions of the other structure.

Pump Case Model

The pump case model is a three-dimensional mathematical representation of the pump casing. The model also includes the cover plate but does not include the cover plate to pump case bolting. Instead, the cover plate is joined rigidly to the upper flange of the pump case. The model's geometry is developed in three phases: The first phase consists of drawing a series of vertical cross sections through the pump case. In the second phase, points (referred to as nodes) are defined on each local coordinate system which form a grid over the geometry of that section. Elements are defined in the last phase of model construction. Two types are used; shells (triangles or quadrilaterals) and solids (wedges or bricks). Triangular shell elements are defined from three nodes while four nodes are required to define the quadrilateral.

Four computer programs, referred to as preprocessors, are used. The preprocessors check the data for undefined nodes, negative nodal areas or volumes (which indicate either duplicated nodes or badly skewed corners), misdefined material codes, and unspecified thicknesses (shell element nodes only).

Element Definitions

The pump case portion of the nodal is built from 746 shell elements and 1,814 solid elements. The cover plate portion consists of 578 solid elements. Shell elements are connected to solid elements by means of 116 constraint sets. A portion of the pump case model is shown in Figure 6.

Method of Analysis (Pump Case)

The three-dimensional finite element method is used to calculate stress levels throughout the pump case. Two computer programs perform the required calculations using as input a three-dimensional mathematical model of the pump case to which specified boundary conditions are applied.

Highly Stressed Areas

Structurally, the pump case is a pressure vessel. As such, volute wall thicknesses and diffuser vane cross sectional areas are for the most part determined by the design pressure. The high stress areas of the pump case for pressure loading (that is, for primary stress levels) are diffuser vanes seven thru nine, the volute wall at hydraulic sections seven thru nine, the junction of the suction nozzle with the lower flange, and the junction of the discharge nozzle with the volute and the flanges.

Closure Model

The finite element model for the closure is an axisymmetric mathematical representation of the components used to close the pressure boundary of the pump casing. The components included in the model are the driver mount lower flange, the heat exchanger, the seal flange, the lower pressure breakdown device, the third seal backup flange, the driver mount to case bolting, and the heat exchanger to driver mount bolting.

The case closure portion of the model consists of 323 axisymmetric ring elements (quadrilaterals or triangles) which are defined by 369 nodal points. The driver mount consists of 174 elements, the case upper flange consists of 40 elements, and the stud and nut consists of 76 elements as shown in Figures 4 and 5.

Method of Analysis - Case Closure

An axisymmetric finite element computer program is employed to

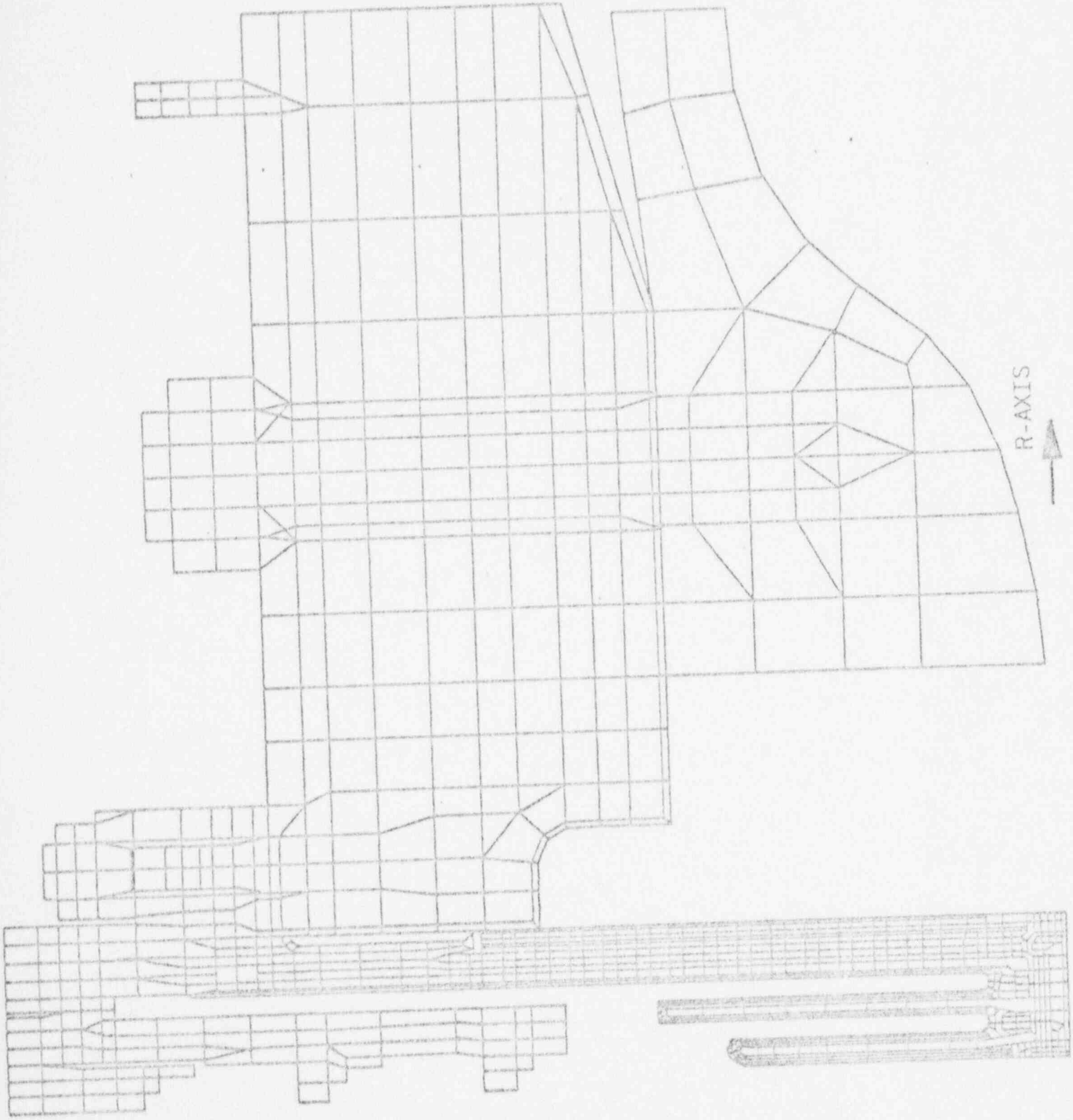


FIGURE 4 FINITE ELEMENT MODEL

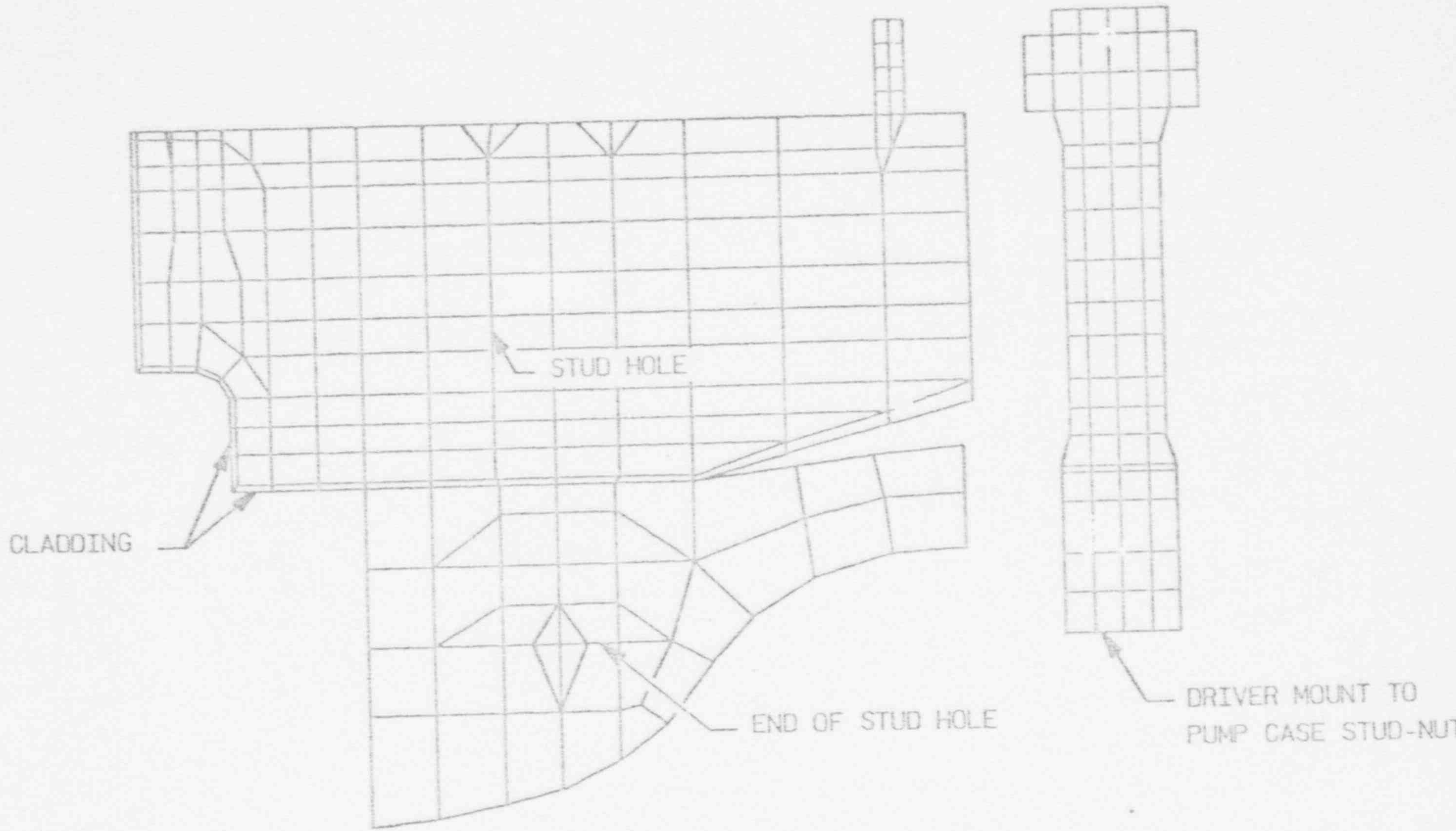


FIGURE 5 EXPLODED VIEW OF CASE CLOSURE MODEL

evaluate the stresses throughout the case closure for internal pressure and bolt preloading. The solution consists of nodal deflections and element coordinate stresses calculated at the element centroid.

Conditions Analyzed

Several conditions are evaluated. These are referred to as Basic Conditions. They include a bolt preload condition, a design pressure condition and a normal operation pressure condition.

STRESS EVALUATION

The Byron Jackson pump evaluated in this report is used in several of the B&W Reactors. A listing of these reactors is given in the Introduction to this report. The peak pressure during an ATWS transient for these reactors has been determined to be slightly less than 3750 psi (see page 3). The allowable stress intensity for ATWS transients are based on ASME Section III rules for Level C conditions, (previously "emergency conditions"). Primary membrane stress is compared to 1.2 S_m (19700 psi at 680°F). Primary membrane plus bending is compared to 1.8 S_m (29500 psi at 680°F). (For additional discussion of allowable stress, see page 2.)

Stress resulting from a pressure of 3750 psi has been compared to allowable values. Calculated stress intensities which exceed allowable values are summarized in Table 3. The number of elements whose stress exceed the allowable are tabulated as a percentage of the elements in the pump case. The stress values are those at the centroid of each element, and may be used as a conservative evaluation of primary membrane stress. (Conservative, because true membrane stress for vanes is the average stress across the section. No attempt was made to remove this conservatism, because stress intensities which include 6 components of stress cannot be averaged directly.)

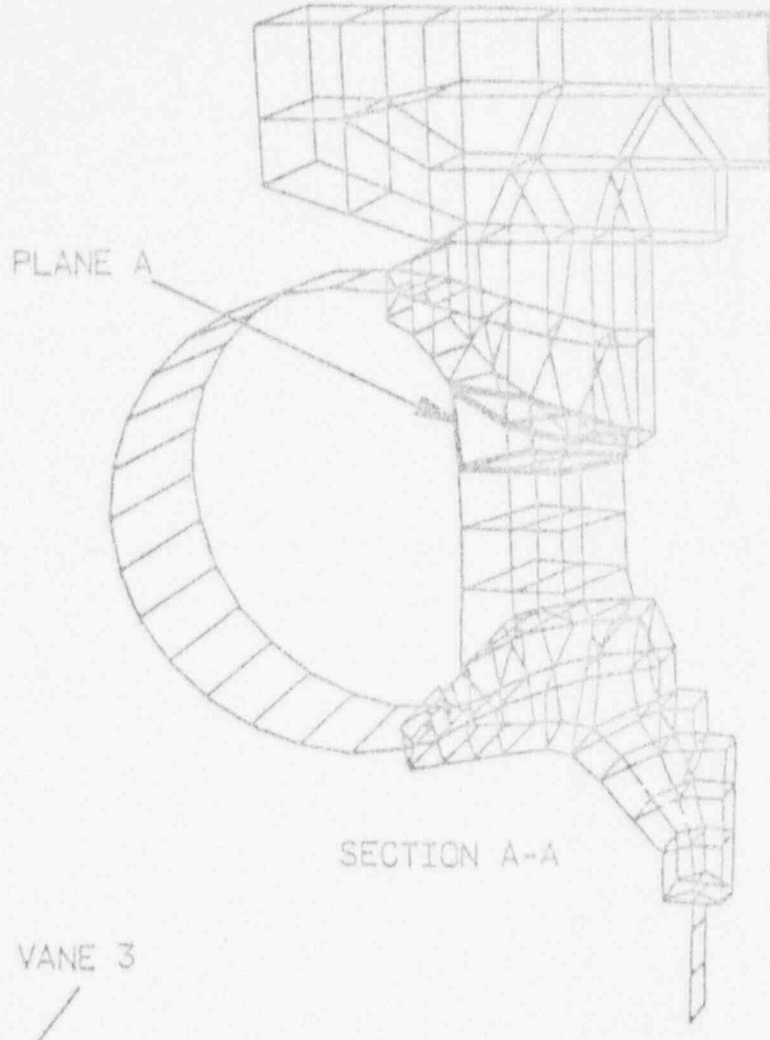
The location of these elements are presented in Figures 6 through 11. The highest stress intensities occur in the root sections of vane Number 8, which is adjacent to the crotch. (Vane 9 is an extension of the crotch.)

CENTROID STRESS INTENSITIES

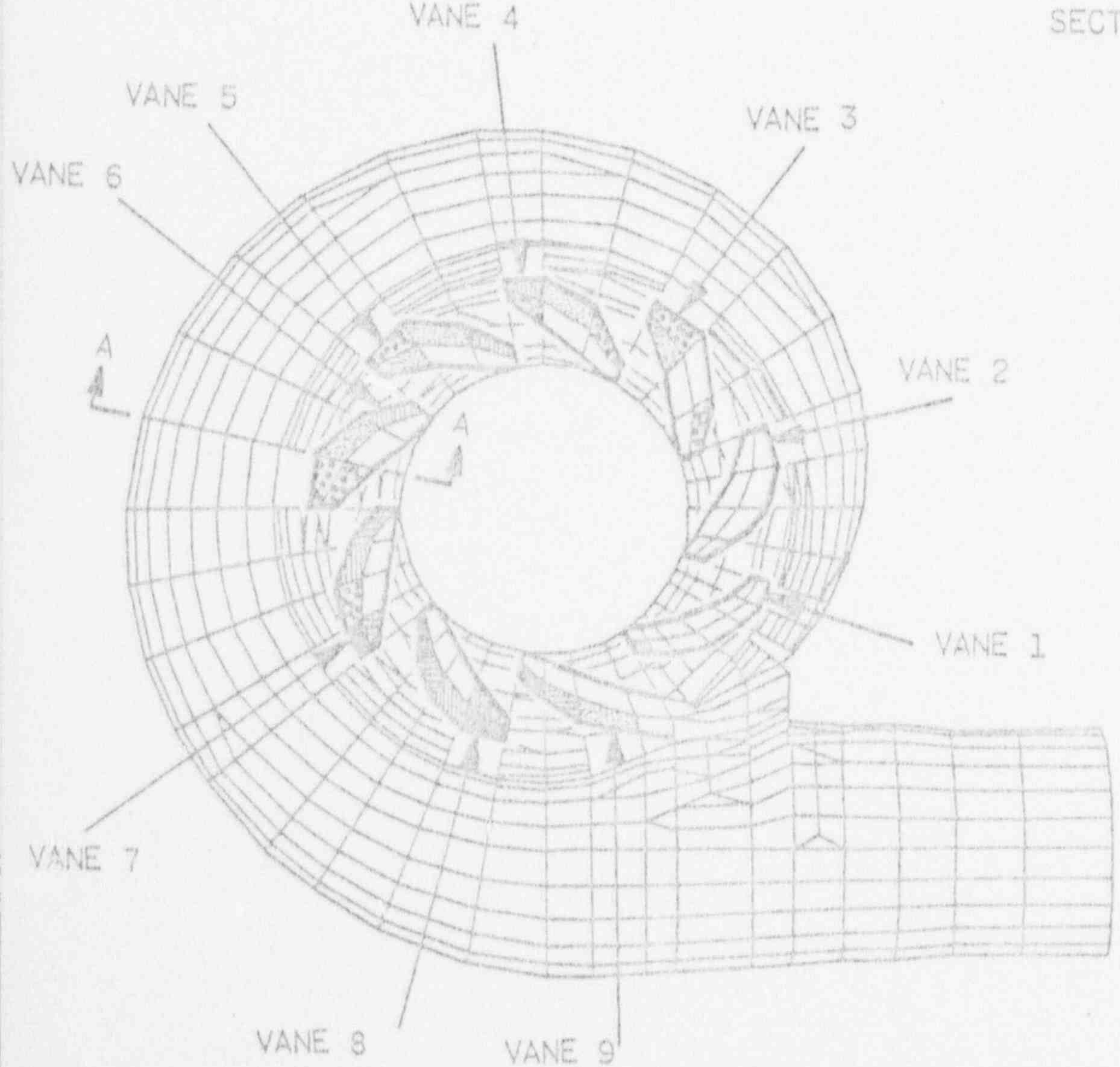
VANES - PLANE A

FIGURE 6

	% 1.2 S_M	% ELEMENTS
	100-110	0.2
	110-120	0.3
	120-130	0.3
	130-140	0.2
	140-150	0.2



SECTION A-A

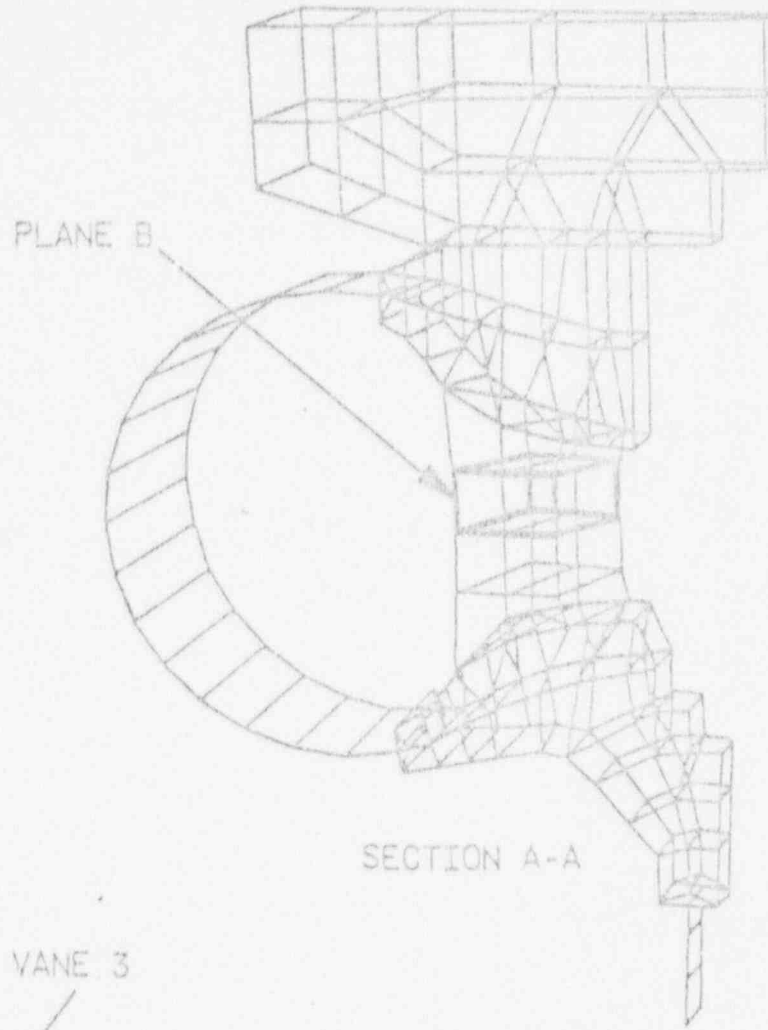


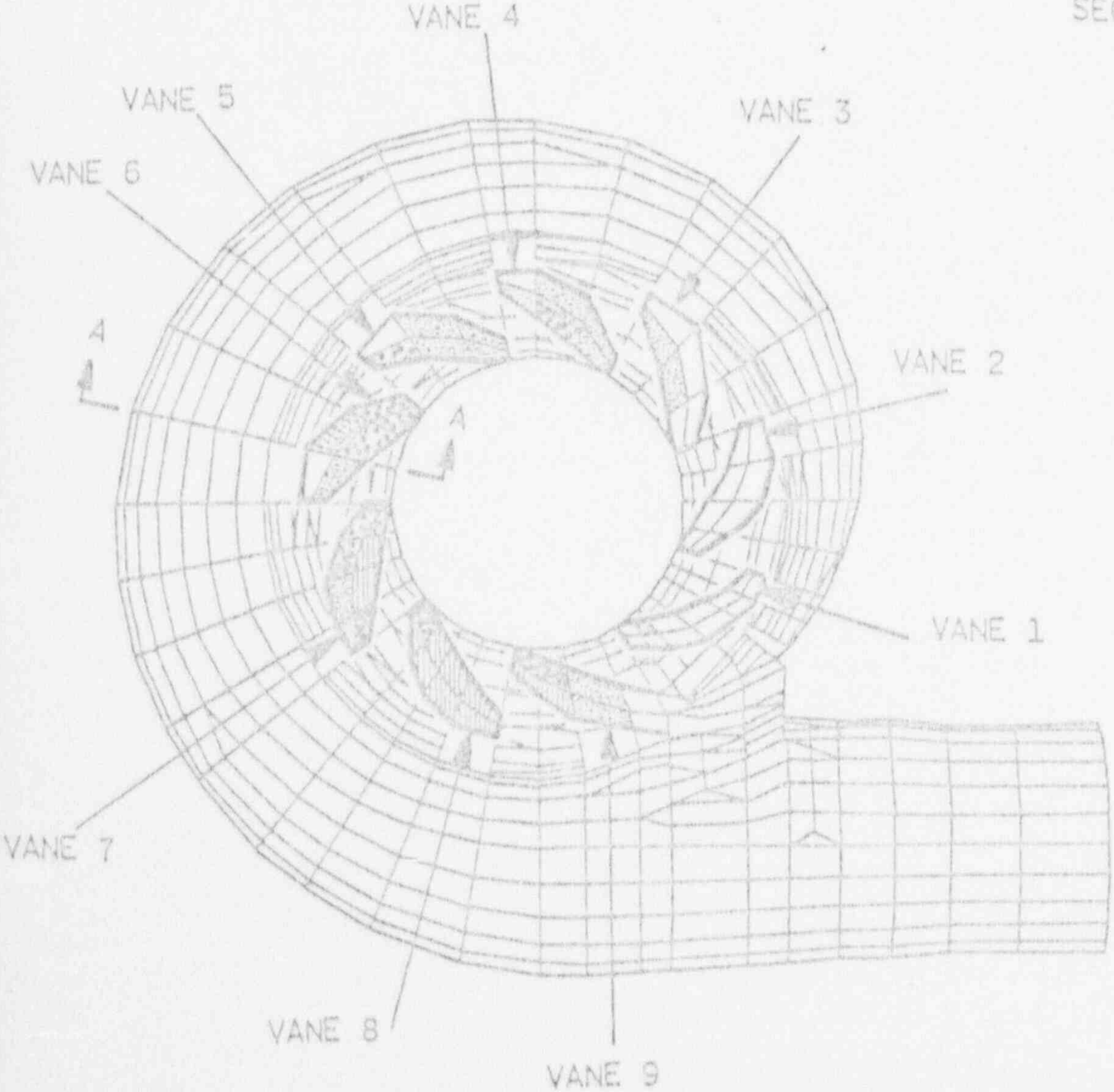
CENTROID STRESS INTENSITIES

VANES - PLANE B

FIGURE 7

	% 1.2 S_M	% ELEMENTS
	100-110	0.6
	110-120	0.5
	120-130	0.4



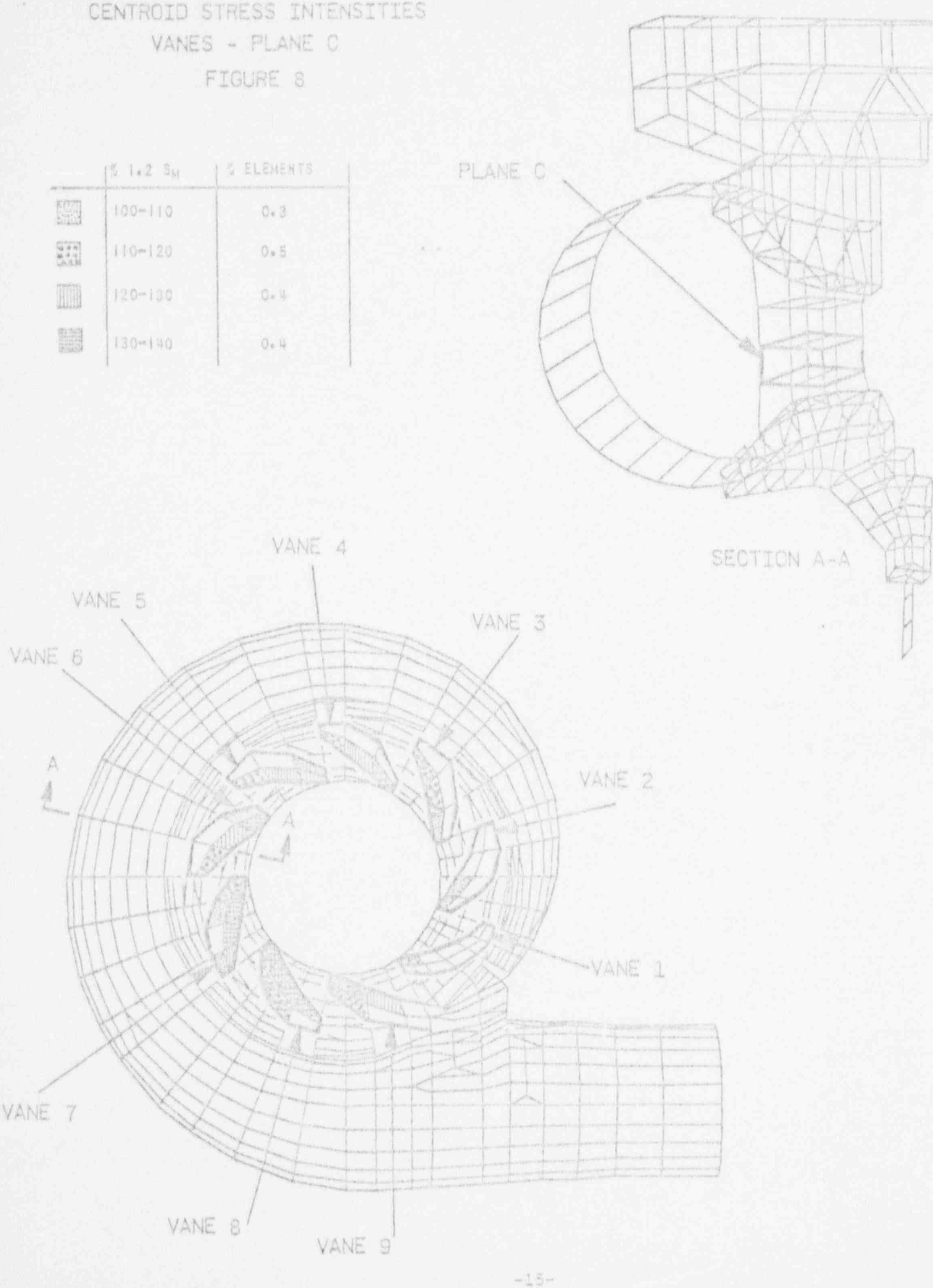


CENTROID STRESS INTENSITIES

VANES - PLANE C

FIGURE 8

	% 1.2 S_M	% ELEMENTS
	100-110	0.3
	110-120	0.5
	120-130	0.4
	130-140	0.4

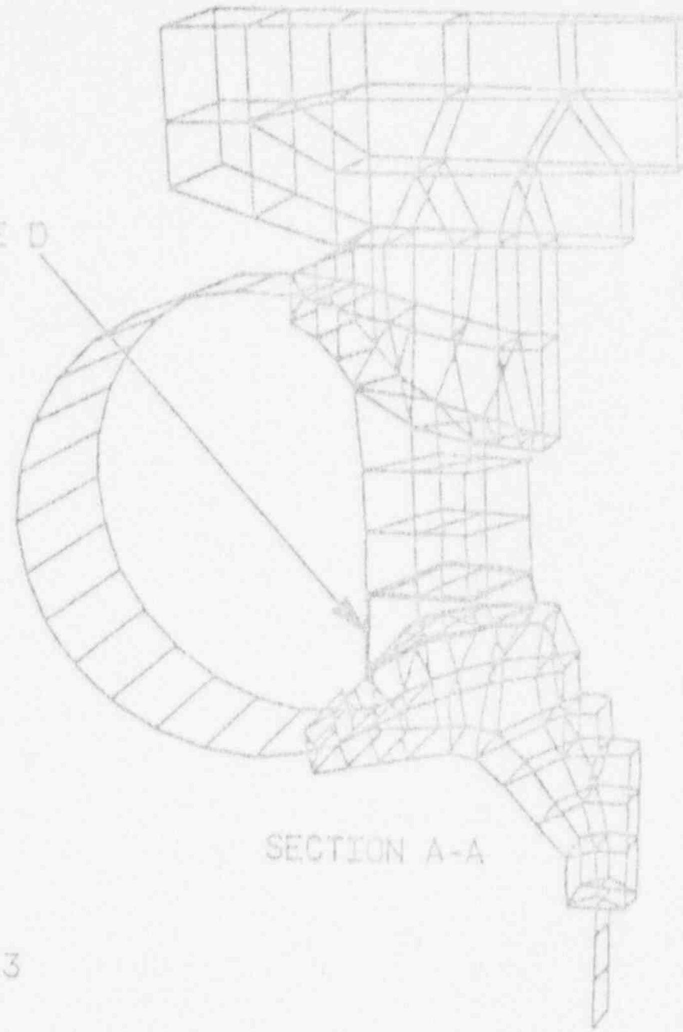


CENTROID STRESS INTENSITIES

VANES - PLANE D

FIGURE 9

	% 1.2 S_M	% ELEMENTS
	100-110	0.4
	110-120	0.2
	120-130	0.2
	130-140	0.2



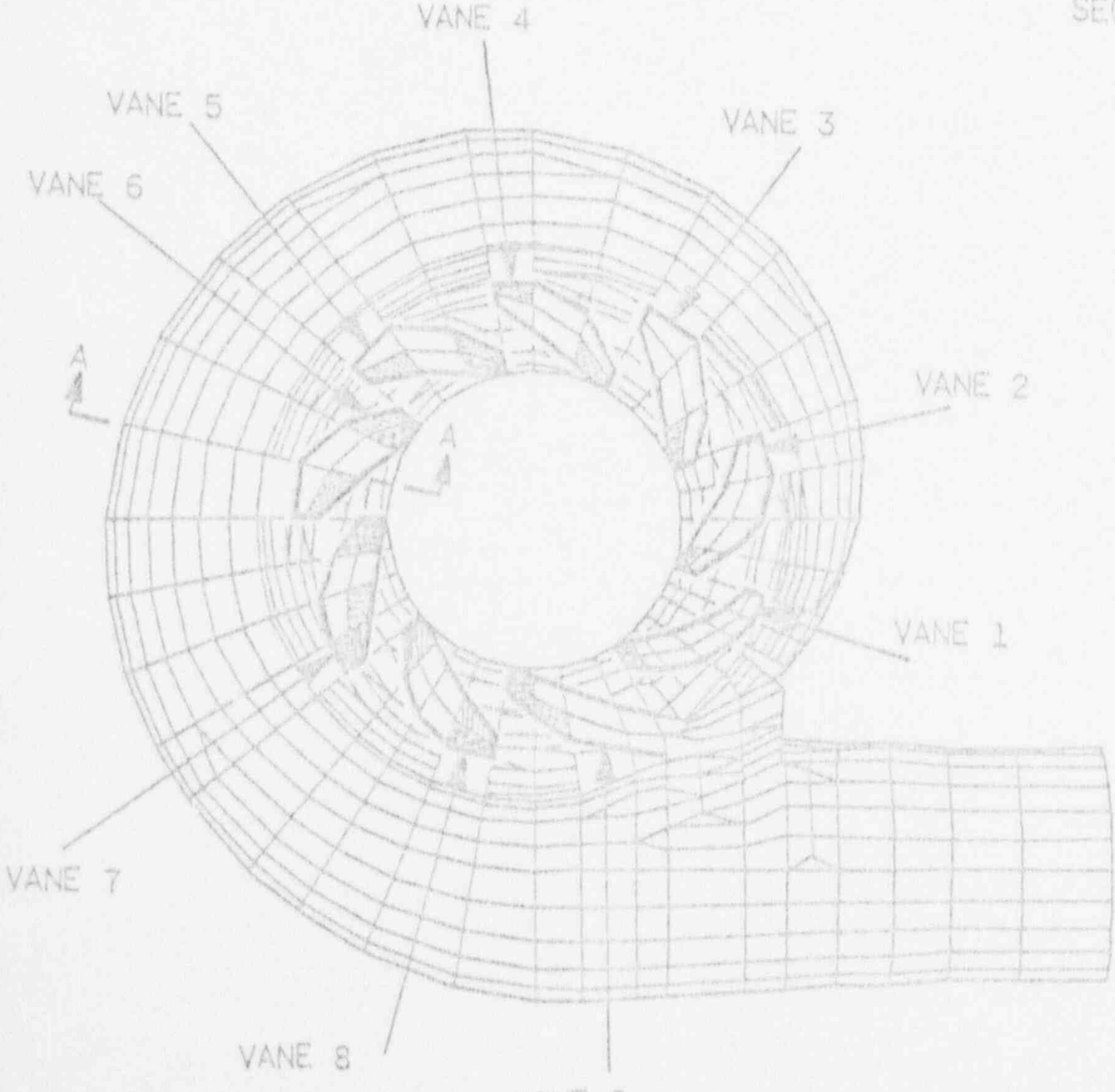


FIGURE 10. CENTROID STRESS INTENSITIES
TOP HALF OF VOLUTE

	% 1.2 S_M	% ELEMENTS
	100-110	3.0
	110-120	1.3

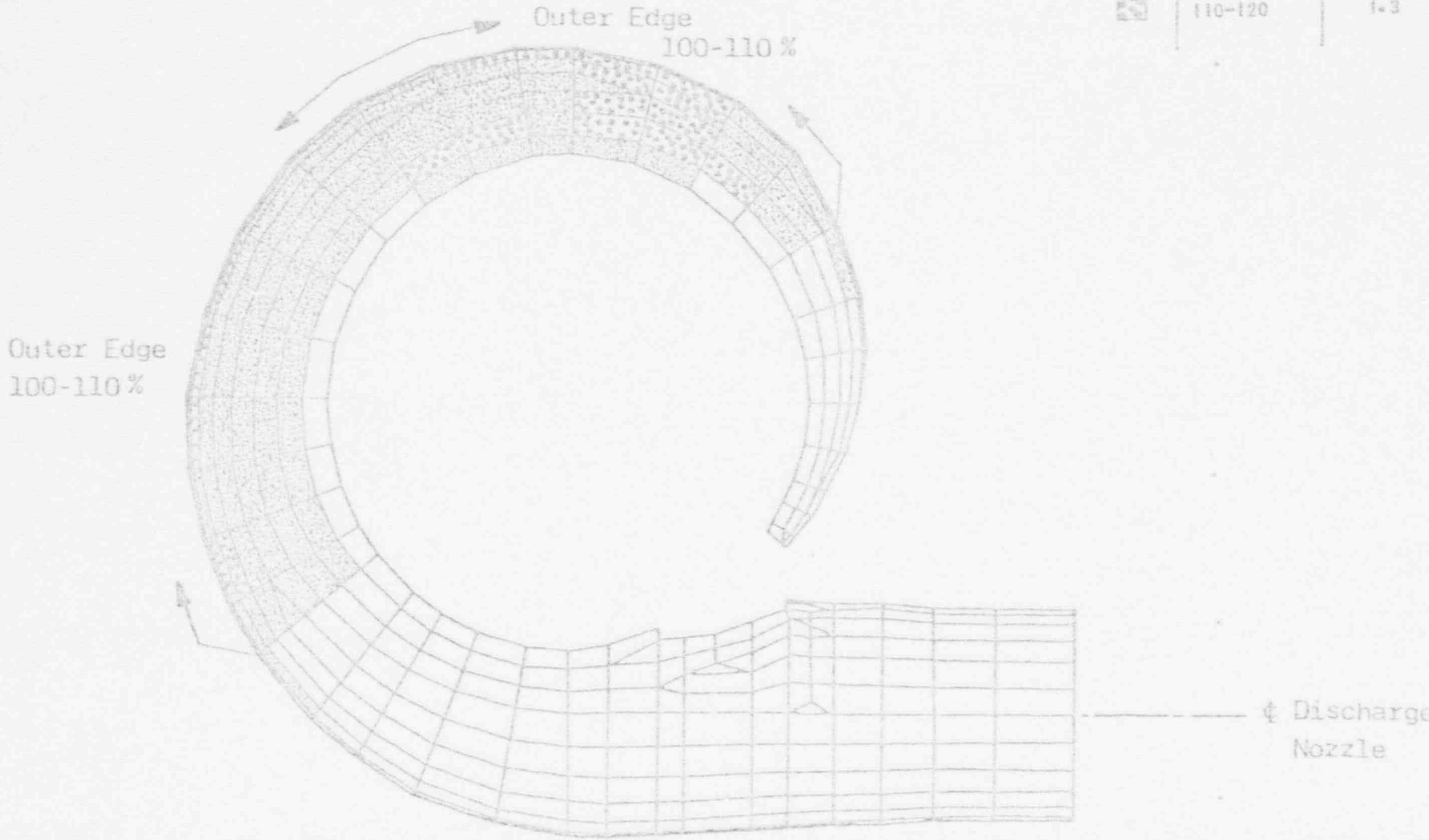
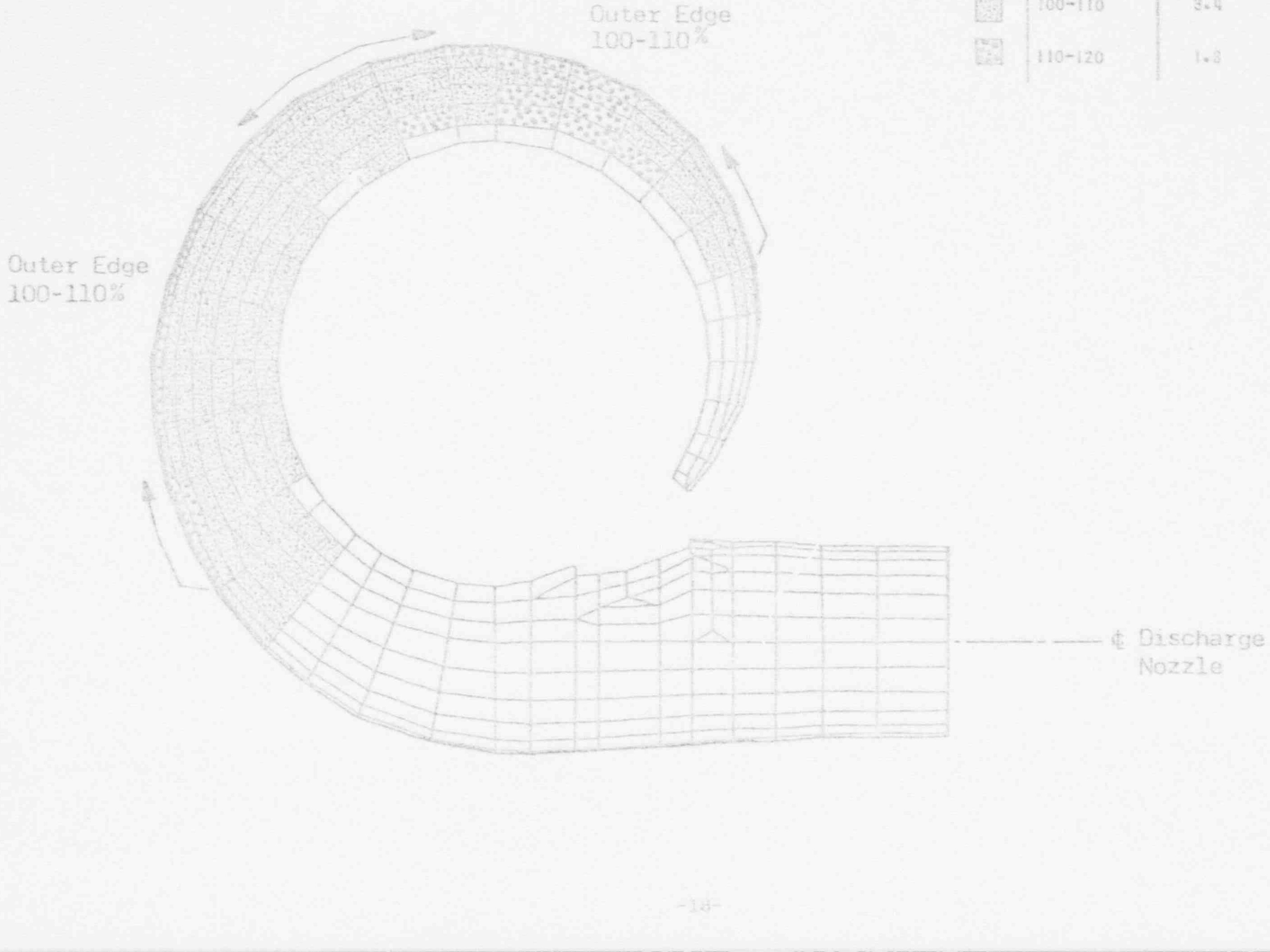


FIGURE 11. CENTROID STRESS INTENSITIES
BOTTOM HALF OF VOLUTE

	% 1.2 S_M	% ELEMENTS
	100-110	3.4
	110-120	1.8



Since the root has larger area than the vane, much of the increased stress in the root elements is believed to be stress which would be classified as peak stress (i.e., due to discontinuities). The membrane stress intensities for the volute shell are less than 15% in excess of the allowable stress. Stress which exceeds allowable values occurs only in the volute shell and in the vanes which connect the inner edge of the volute. The flanges, closure plate, and closure bolts which surround the rotating machinery are stressed below allowable values.

The combination of primary membrane plus primary bending stress was evaluated by comparing stress intensities at the surfaces of the elements with Level C allowable stress (i.e., 1.8 S_m). It was found that less than 1% of the elements had surface stresses which exceed this value, and that these were located at the root of the vanes. It is believed that a detailed evaluation would show that most of the stress which exceeds allowable stress in these few elements (i.e., 25) is attributable to peak stress rather than primary bending.

TABLE 3 - SUMMARY - QUANTITY OF ELEMENTS IN PUMP CASE WHICH EXCEED 1.2 S_m

% 1.2 S_m	% of Elements
100-110	7.9
110-120	4.1
120-130	1.3
130-140	0.8
140-150	0.2

Closure Bolts

Two factors are of interest for evaluation of closure bolts. The bolts must be strong enough to withstand pressure and gasket loads, and the bolts should preload the gaskets to seat them for the evaluated pressure.

A. Bolt Strength

The bolts are made from ASME SA 540 Grade B23 Class 4 material. The stress value S_m for this material is 32100 psi at 650°F. Taking the ratio of actual bolt area to the required bolt area for the design pressure of 2500 psi and applying the applicable allowable stress rates for bolt sizing and service conditions for the ATWS postulated pressure of 3750 psi, it is found that the resulting bolt stress is 39% of the allowable stress.

$$\frac{\sigma}{2S_m} = \frac{138.2\ in^2}{264.8\ in^2} \times \frac{3750\ psi}{2500\ psi} \times \frac{1\ S_m}{2S_m}\ (100\%) = 39.4$$

It is concluded that the bolt has adequate strength to withstand the postulated ATWS transient.

B. Gasket Seating

The bolts for pumps at different reactor sites are preloaded to different values. The minimum preload is 25280 psi. By ratioing the bolt stress to seat the gasket *at 2500 psi to the postulated ATWS transient of 3750 psi, it is found that a bolt stress of 25130 psi is required to seat the gasket. It is concluded that the bolts are sufficiently preloaded to seat the gasket, and that the closure will not leak during the transient. Additional margin exists for long-term leakage because the ATWS transient pressure is of short duration.

Heat Exchanger and Seal Closure Bolts

A similar evaluation of these bolts was made. It was found that bolt sizing was adequate to maintain stress below ASME allowable stress, and that sufficient preload is applied to maintain required gasket seating loads at the 3750 psi ATWS transient pressure.

* Determined using Appendix E 1210(a) (i) of ASME Pressure Vessel Code, Subsubsection of Section III.

PUMP CASE DISTORTION

The magnitude and distribution of the stress at 3750 psi pressure was given in the preceding section. A related consideration is whether the distortion associated with this stress field would affect the ability of the impeller to rotate. This effect has been found to reduce the clearance between the impeller and the casing by only 5% (5 mils in 96 mils). It is concluded that impeller rotation will not be affected by this amount of pump case distortion. This evaluation is discussed in detail below.

Methodology

The clearance between the impeller and the pump casing is smallest at the wear ring in the suction "flange". The minimum diametral clearance at this point is .096 inches as shown in Figure 2. The pump flanges at the suction and at the pump closure are quite massive, and the stresses in this region are within allowable stress. These therefore behave as two rigid disks, connected by the vanes. The average strain in the direction parallel to the pump centerline is determined for each vane. These values are given in Table 4. From this data, the change in slope between the two flanges is determined. This, in turn, permits the lateral motion of the wear ring to be calculated, assuming the suction flange rotates about a point within the plane of the vanes, causing a lateral displacement of the wear ring, which is below the "hinge point".

This value, which is based on the elastic analysis, is increased by an inelastic factor which is the ratio of a secant modulus to the elastic modulus. The secant modulus is based on a stress-strain curve which uses ASME minimum stress values. It is assumed that all vanes have the reduced modulus determined for the highest stressed vane.

TABLE 4 - AVERAGE AXIAL STRAIN IN VANES

Vane	Average Strain
1	5.4×10^{-4}
2	5.6×10^{-4}
3	6.6×10^{-4}
4	7.3×10^{-4}
5	7.4×10^{-4}
6	7.5×10^{-4}
7	8.3×10^{-4}
8	8.7×10^{-4}
9	7.8×10^{-4}

Average strain, 9 Vanes

$$\bar{e} = 6.7 \times 10^{-4} \text{ in/in}$$

Determination of Secant Modulus

Maximum equivalent stress, vane 8

$$\sigma^*_{max} = Ee = 24.9 \times 10^6 \, (8.7 \times 10^{-4}) = 21,650 \text{ psi}$$

Total strain at 21650 psi stress

$$e_{secant} = .0053 \text{ (see Figure 12)}$$

Secant Modulus

$$E_s = \frac{\sigma}{e_s} = 21650 = 4.1 \times 10^6$$

Inelastic ratio

$$\frac{E}{E_s} = \frac{24.9 \times 10^6}{4.1 \times 10^6} = 6.1$$

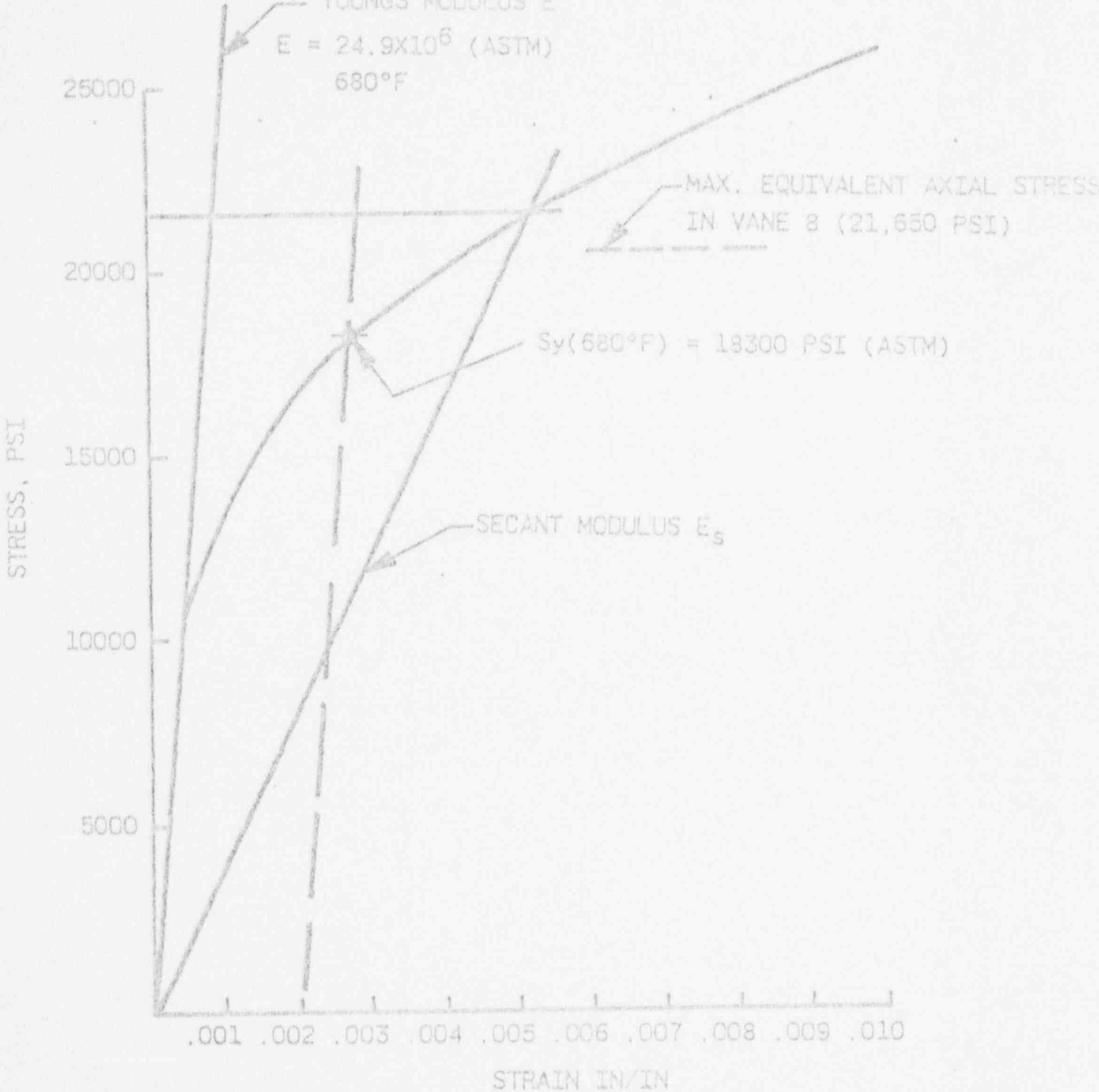


FIGURE 12 DETERMINATION OF INELASTIC STRAIN RATIO

Wear Ring Calculation

The effect of strain in the vanes on the shift of the suction flange and wear ring is calculated as follows:

The tilt of the suction flange with respect to the closure flange is determined by first determining the axis which remains parallel to both flanges. This is calculated by summing the product of the difference between the overall average strain and the strain of each vane, by the coordinate Y for each vane and setting this sum of moments to zero. Stated mathematically;

$\sum_1^9 (e_i - \bar{e})\, Y_i = 0$ $\qquad$ e_i = average strain in each vane

$\bar{e}$ = average strain in all 9 vanes

where $Y_i = R \sin \beta_i$; $\beta_i = \alpha_i - \gamma$; $\alpha_i = 20^\circ + 40\ (i-1)$

This is solved by iteration, using the midpoint between vanes 1 and 9 as an arbitrary origin. Several angles γ are used as trial values until the summation converges to zero. This angle was determined to be 70.7 degrees. This is the axis about which the moments are balanced, and which does not rotate. In other words (see Figure 13), the flanges near vanes 4 and 9 remain equidistant.

The maximum slope occurs perpendicular to this axis (i.e., from vanes 7 to 2). The value of the slope is determined in a similar manner, except that the value of the slope is determined such that the product of strain differences and the coordinate X is equal to zero. Stated mathematically;

$$\sum_1^9 \left[(e_i - \bar{e}) - \dot{e} X_i\right] X_i = 0$$

where $X_i = R \cos \beta_i$

$\beta_i = \alpha_i - \gamma$; $(\gamma = 70.7^\circ)$

$\alpha_i = 20^\circ + 40^\circ\ (i-1)$

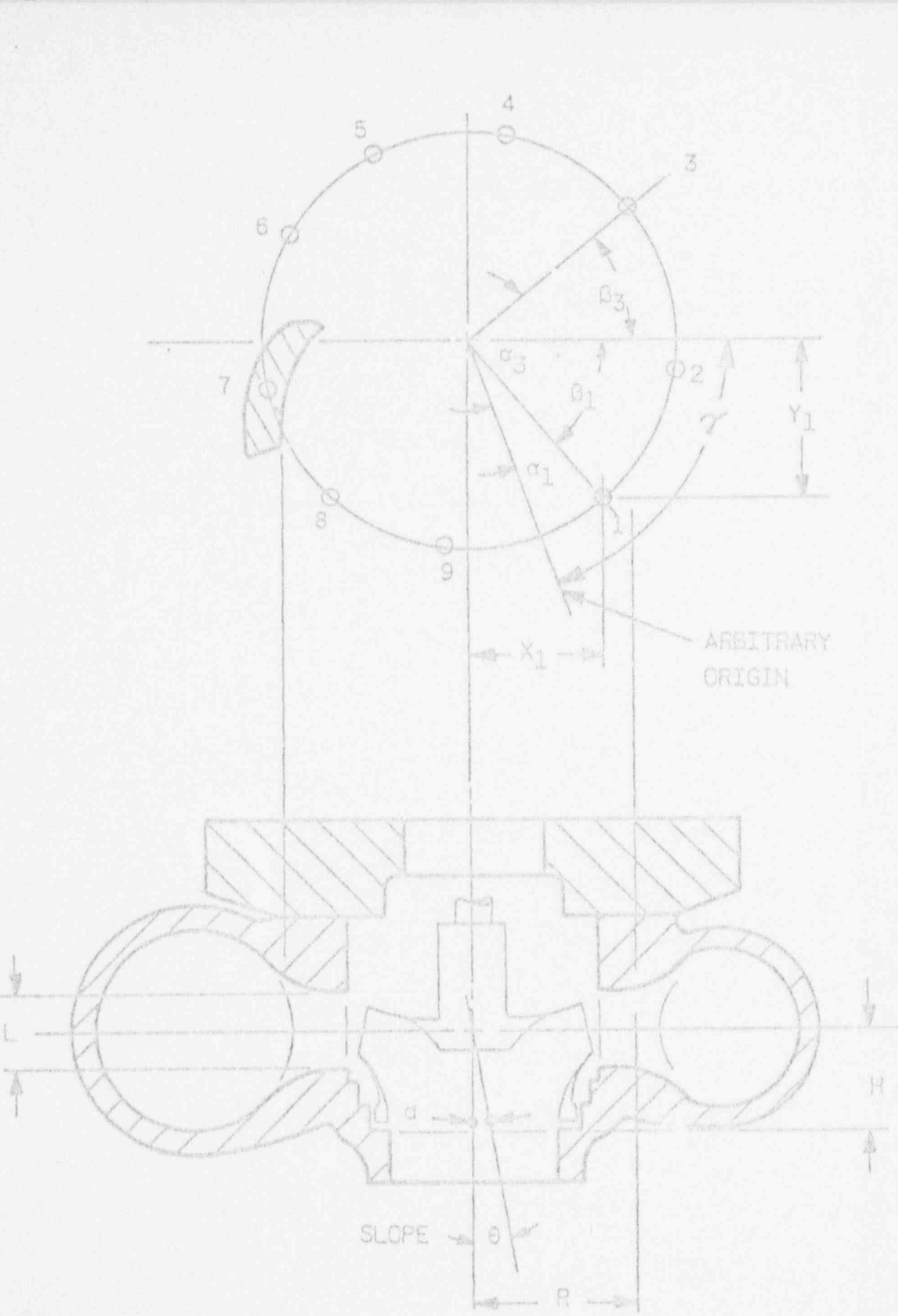


FIGURE 13 DISTORTION MODEL

$\dot{\theta}$ = equivalent strain rate for slope θ (i.e., increased strain from pump centerline to the vane)

ϵ_i = average strain in each vane

$\bar{\epsilon}$ = average strain in all 9 vanes

By iterative trials for the differential strain rate $\dot{\epsilon}$, the summation goes to zero at $\dot{\epsilon} = .045 \times 10^{-4}$ per inch. The slope is then

$$\theta = \frac{\epsilon_{max} L}{R} = \dot{\epsilon} L$$

Where

L = Vane Length, = 12 inches

and $\epsilon_{max} = \dot{\epsilon} R$

Numerically

$\theta = 0.045 \times 10^{-4}\ (12) = 5.4 \times 10^{-5}$ radian

and the displacement

$d = \theta H = 5.4 \times 10^{-5} (15) = .0008$ inches

where H = distance from vanes to wear ring, 15 inches.

Applying the inelastic factor

$\frac{E}{E_s} = 6.1$ (see page 22)

The wear ring displacement

$$d_{inelastic} = d_{elastic} \left(\frac{E}{E_s} \right)$$

$d_{inelastic} = .0008\ (6.1) = .005$ inches (This also includes elastic recovery displacement)

This is small compared to the minimum diametral clearance of .096 inches; therefore, it is concluded that distortion of the vanes will not produce suction flange displacement which would cause interference with the impeller rotation.

STRUCTURAL INTEGRITY

In the preceding evaluations, the distribution of stress which exceeds ASME allowable stress for Level C events has been presented for the pump case and the diffuser vanes. Stress in all closure bolts are within allowable stress, and gasket seating has been found to be sufficient to meet ASME requirements at the ATWS pressure. A quantitative estimate of the effect of pump case distortion indicated that impeller rotation will not be restrained.

Several of the most significant stress levels are presented on a stress-strain curve for the pump material (see Figure 14). The purpose of this graphic display of stress values is to illustrate that the stress values are significantly lower than ultimate strength, and that the resulting strain is small compared to the material elongation capability.

The curve was constructed to pass through the ASME design allowable values at 680°F (e.g., yield strength at 0.2% offset, and the ultimate strength). Typical material values are expected to be higher.

For the diffuser vanes, two stress values are shown on the curve. The maximum stress in vane 8 exists at the base of the vane, and undoubtedly includes stress which is classed by the ASME as peak stress. (Peak stress is not required to be included for Level C events.) This value represents the upper bound for stress at the centroid of any element. The other value is the average equivalent axial stress for vane 8. It is based on the axial strain components averaged across a horizontal plane, and represents the maximum equivalent membrane stress in the diffuser vanes (see also page 22).

The maximum stress in the volute is also shown. The combination of primary membrane plus bending stress is within ASME allowable stress. The value shown is the primary membrane stress, and is taken from the centroid

stress for the shell-type elements.

It is concluded that the structural integrity of the pump is adequate to maintain the pressure boundary intact for the 3750 psi pressure postulated for an ATWS incident.

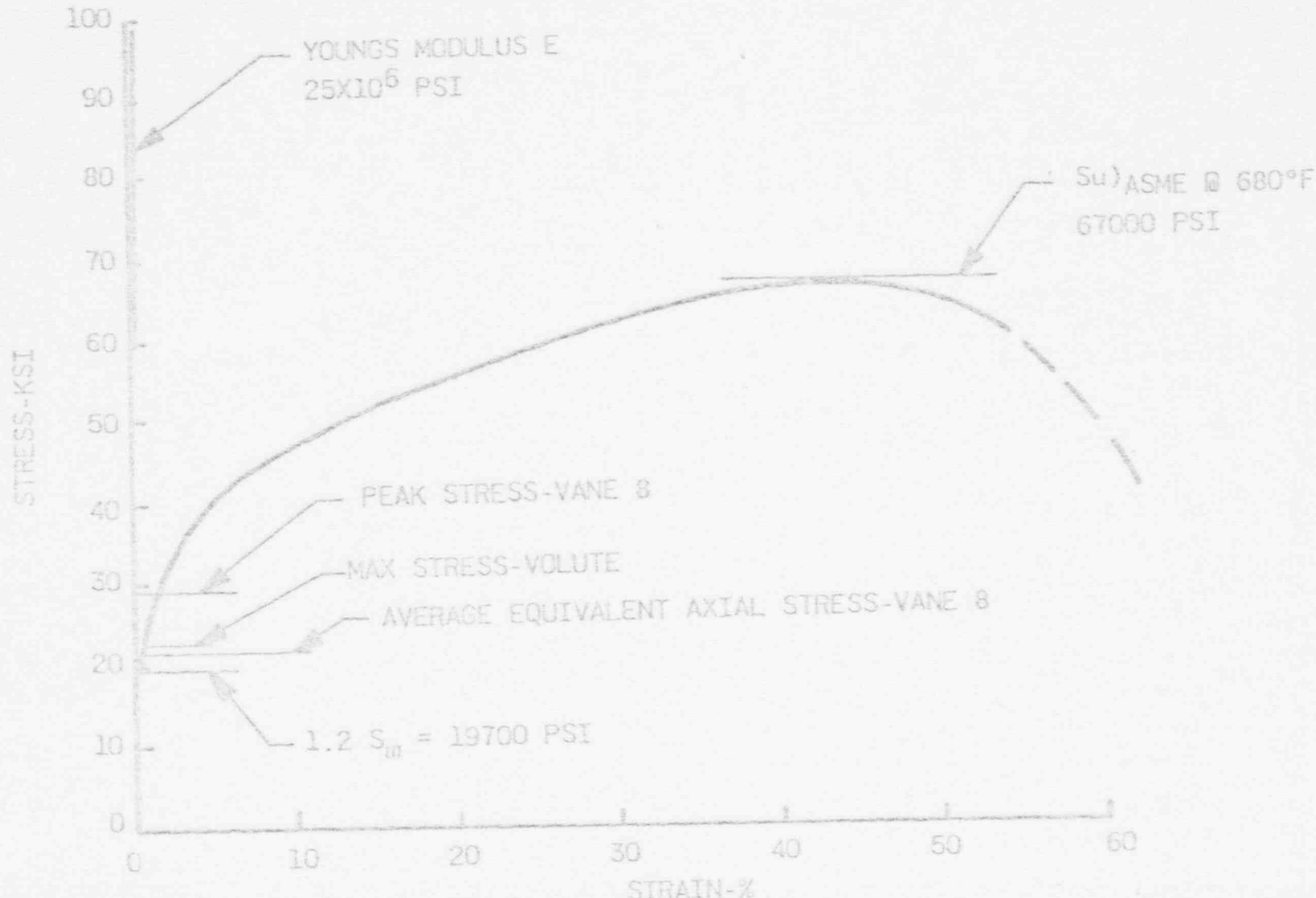


FIGURE 14 STRESS COMPARED TO STRESS-STRAIN CURVE