To the Graduate Council:

I am submitting herewith a thesis written by Haren Sheth entitled "Stress Analysis of Support Design for Horizontal Pressure Vessels Subjected to Pressure and Seismic Loads." I recommend that it be accepted in partial fulfillment of the requirements for the degree of Master of Science, with a major in Engineering Science.

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STRESS ANALYSIS OF SUPPORT DESIGN FOR HORIZONTAL PRESSURE VESSELS SUBJECTED TO PRESSURE AND SEISMIC LOADS

A Thesis

Presented for the

Master of Science

Degree

The University of Tennessee, Knoxville

Haren Sheth

December 1976

1306050

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ABSTRACT

The purpose of this study is to present the results of a study in which the author has used the finite element method to compare displacements, interface pressure and stresses in the saddle-shell interface region for a particular pressure vessel for cases where the shell is both welded and unwelded to the saddle supports, and subjected to the vertical loads. In addition, support designs for several horizontal pressure vessels were analyzed when they were subjected to seismic loads. The results of this investigation are compared with the results of several previous theoretical and empirical pressure vessel studies.

The finite element results show that when the pressure vessels were subjected to vertical loads only, the maximum stresses obtained are markedly less than some recently published results but agree favorably with some standard codified design procedures. When these vessels were subjected to seismic loads, the maximum stresses in the saddle structure were extremely high. This could cause failure of the support due to local buckling. These results also indicate that the development and analysis of saddle designs capable of resisting seismic loading should be the next stage of research effort in this area.

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LIST OF SYMBOLS

A	distance from tangent line of the head to the center of saddle support
ah	horizontal component of the acceleration
B	base plate length
b	saddle width
CR	center rib
Ds	density of steel
Dw	density of water
Fh	horizontal force
g	gravitational constant
h	saddle height
IR	inner rib
k	width of wear plate
L	tangent to tangent length
m	mass
n	number of ribs
OR	outer rib
P	internal pressure
Q	resultant load on one saddle
R	radius of the cylindrical shell
r	rib width
S_a	allowable stress of the material
S_{bx}	longitudinal bending stress
$S_{b\phi}$	circumferential bending stress

S_{mx}	longitudinal membrane stress
$S_{m\phi}$	circumferential membrane stress
S_{my}	circumferential membrane stress in ribs
t_l	thickness of web
t_r	thickness of ribs
t_s	thickness of shell
t_w	thickness of wear plate
W_s	weight of shell
W_w	weight of water

CHAPTER I

INTRODUCTION

Horizontal pressure vessels for liquid storage have found many applications in industry. When such vessels were made to carry their own weight and to undergo internal pressure, many stress analysis problems appeared at the junction of the shell and its supports. There have been very few investigations made towards the study of these problems.

In 1941, Hartenburg (1) examined several saddle supported, unstiffened horizontal pressure vessels and came to the conclusion that critical stresses in the shell were induced by the saddle tips due to concentrated loads existing at these locations. The worst conditions occurred when the vessels were completely full of liquid. When the saddle angle was reduced, the critical stresses were increased. These stresses were, however, independent of the saddle width. Hartenburg also concluded that the simple beam theory did not give satisfactory stress values in the pressure vessels near the saddle-shell interface.

Wilson and Olson (2) performed experiments on the unstiffened and ring stiffened steel cylinders supported by a centrally loaded saddle and confirmed the results obtained by Hartenburg (1). They assumed that the simple beam theory was adequate in determining the longitudinal bending and tangential shear stresses. Their experimental results for ring stiffened cylinders agreed quite well with the simple beam analysis. However, the theory proved to be highly inadequate for unstiffened cylinders.

In 1951 L. P. Zick (3) published a paper to predict the approximate stresses that exist in cylindrical vessels supported on two saddles at various locations. The American Society of Mechanical Engineers (A.S.M.E.) has presented Zick's analysis as a recommended practice in designing horizontal pressure vessels. The British Standard 1515 also refer to his analysis as a design criteria.

Recent papers by Lakis and Dore (4), and Wilson and Tooth (5) have presented an analytical method capable of predicting the displacements, saddle-vessel interface pressure and stress resultants throughout the vessel. Both papers have presented their results for a particular gometric configuration of the shell-saddle in which the supports were assumed to be rigid and not fastened to the vessel. However, because a large number of pressure vessels constructed today must be designed to withstand seismic loads, the supports must be securely fastened to the vessel. This is usually accomplished by welding the supports to the shell. A literature survey has revealed that there is no codified design procedure available analogous to Zick's, which can readily predict the stresses at the saddle-shell interface for the case when the saddle is welded to the vessel. The analyses presented by references (3), (4), and (5) do not concern themselves with the seismic design of horizontal vessels. The question arises whether these analyses are adequate to incorporate the state of stress at the saddle-shell interface, when the vessel is subjected to horizontal as well as pressure loads.

When the pressure vessel is welded to the saddle support, the support is required to transfer any horizontal load to the support

anchors and must, therefore, be designed to sustain this loading. The American Society of Mechanical Engineers (A.S.M.E.) Pressure Vessel Code requires that the support be built according to sound engineering principles; and since pressure vessels are essential to the safety and continuity of operations in such critical national industries as petroleum processing plants and public utilities, including nuclear power plants, it is certainly imperative that supports be properly designed to ensure that the function of the pressure vessel not be impaired because of a seismic disturbance. Currently there is no specific analysis available to preduct the stresses in the saddle structure. Consequently, in attempting to design a welded saddle support capable of resisting a specified horizontal load, an engineer is forced to rely primarily on his own former design experience. Since there are many engineers with varying analytical abilities engaged in the design of pressure vessels and their supports, these engineers will turn out a variety of designs for each pressure vessel-saddle support configuration. Because there are no standard design guidelines to follow, it is difficult to verify any of their designs; and because of the complexity of the problems involved, it is likely that some of these designs will be inadequate.

The purpose of this investigation is two-fold; first, several typical pressure vessel designs are analyzed for stresses in the saddle-shell interface region to determine the adequacy of Zick's analysis for the case where the saddle is welded to the pressure vessel. Second, several saddle supports structures are analyzed for stresses when subjected to horizontal loads to determine the adequacy of the support design.

CHAPTER II

SCOPE OF WORK

In this investigation, the finite element method is used to compute the stresses and displacements throughout the vessel and its supports.

In the Part 1, the stresses and displacements throughout the shell as well as the interface pressure distribution are calculated. These are then compared with the results of references (3), (4), and (5) for both welded and unwelded saddles.

In the Part 2 of this investigation, preliminary stress analysis is performed to:

i) determine the maximum stresses and their locations in the saddle-shell interface region as a function of saddle angle and height of the saddle.

ii) compute the principle stresses induced at the ribs.

iii) investigate anchor bolt behaviour due to horizontal loads.

iv) compare the results of this study with those of reference (3).

PART I

STRESS ANALYSIS OF A TWO-SADDLE SUPPORTED HORIZONTAL PRESSURE VESSEL

CHAPTER III

THE SUPPORT OF PRESSURIZED VESSELS

I. SOLUTION PROCEDURE

Using the finite element method and the SAP IV computer program (6), the state of stress, displacements and interface pressure distribution for a saddle supported, cylindrical pressure vessel in the vicinity of the saddle-shell junction was computed. Figure 1 shows the geometric configuration of the pressure vessel and its supports, and Figure 2 shows the finite element mesh used. To take advantage of symmetry, only one quarter of the vessel was modeled. The heads of the vessel were replaced by beam elements with small torsional stiffness and large axial and bending stiffness connecting the node points along the circumference of the cylinder at the locatin where the heads would normally be welded to the cylinder. The axial force in the cylinder caused by pressure acting on the heads was simulated by applying the appropriate concentrated longitudinal loads at these nodes. These approximate boundary conditions were used because they very nearly matched those boundary conditions used in references (4) and (5).

Two design situations were considered in this investigation. First, the saddle support was assumed rigid and not fastened to the vessel; and second, the support was assumed rigid and welded to the vessel. The present work presents the analysis of a pressure vessel with the same geometry as the one analyzed in references (4) and (5), and with the following boundary conditions at the saddle-shell interface.

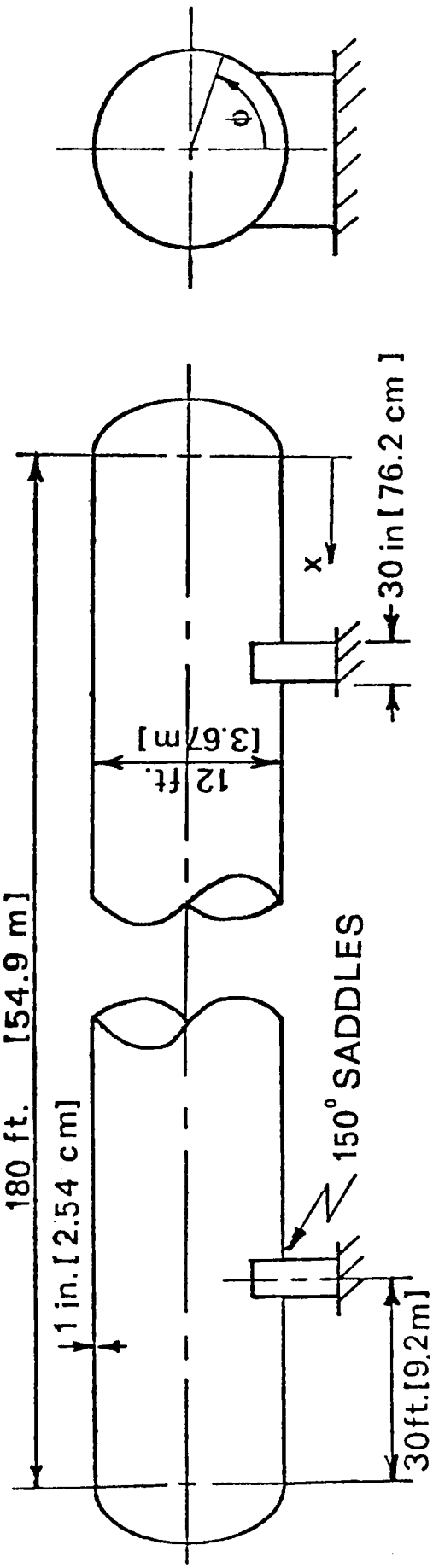

Figure 1. Vessel Dimensions

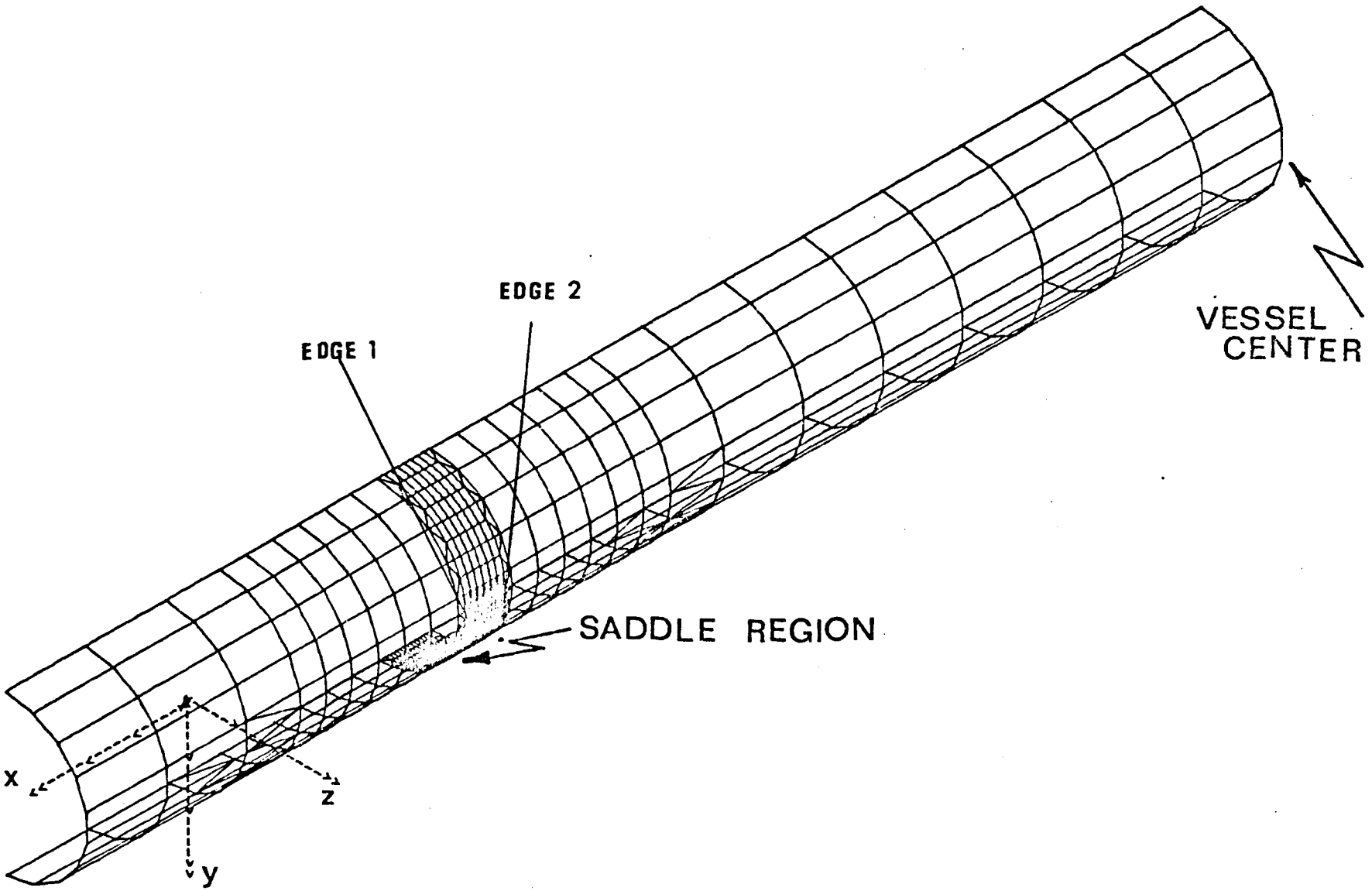

FIGURE 2 . FINITE ELEMENT MODEL

For the unwelded case, the vessel was analyzed by permitting the saddle-shell interface pressure to vary in both the circumferential and longitudinal directions. This was accomplished by using infinitely stiff axial boundary elements normal to the vessel in the saddle-shell juncture region which were capable of resisting only compressive loads. The numerical procedure to determine the pressure distribution was, of course, an iterative process. As in reference (5), for the case with the saddle welded to the vessel, normal displacements for all nodes in the saddle-shell juncture region were restricted. In addition, the longitudinal and circumferential nodal displacements were restricted along the perimeter of the interface region.

For the numerical results given, the vessel was assumed to be made from steel with a Young's modulus of 29×10^6 psi (2×10^{11} Pa), Poisson's ratio of 0.3 and a density of 2.84×10^{-1} lb/in^3 (7.695×10^4 n/m^3). Also the vessel was assumed filled with a liquid of density of 3.611×10^{-2} lb/in^3 (9.783×10^3 n/m^3). The weight of the vessel was considered in all calculations.

II. RESULTS AND DISCUSSION

For future reference, the circumferential lines on the vessel which pass through the edges of the saddle toward the center of the vessel and toward the head of the vessel are denoted as Edge 1 and Edge 2, respectively. The circumferential line passing through the center of the saddle is denoted as the Saddle Center. As shown in Figure 2, the boundary at the center of the vessel is denoted as the Vessel Center. The vessel was assumed to be subjected to pressure loads and dead weight of the cylinder.

Tables I and II show the pressure distribution at the saddle-shell junction for the welded and unwelded cases, respectively. The higher contact pressures occur along Edge 1 with the highest pressure occurring at the tip of the saddle (ϕ = 75 degrees) along Edge 1. As can be seen from these tables, the pressure distribution is not constant in the longitudinal direction as assumed in references (4) and (5). However, a comparison of Tables I, II, and III shows that the general shape of the pressure distribution in the circumferential direction somewhat approximates that given in these references.

Figures 3, 4, and 5 compare the radial and tangential displacements for the present finite element results with those of references (4) and (5) at the Saddle Center and Vessel Center. The radial displacements are considered positive in the direction of the outward normal to the shell. The tangential displacements are considered positive in the direction of increasing ϕ as shown in Figure 1. The graphs show that the displacements agree closely with those of references (4) and (5) for the welded and unwelded case except for the radial displacements of the Vessel Center shown in Figure 4.

Figures 6 through 13 illustrate the stress resultants in the pressure vessel for both the welded and unwelded cases. The stresses are given at Edge 1, Edge 2, and the Saddle Center for the present investigation, but only at the Saddle Center for references (4) and (5) since these references only had the stresses tabulated at this location. N_x and N_ϕ are longitudinal and circumferential membrane stress resultants, respectively. M_x and M_ϕ are longitudinal and circumferential bending stress resultants, respectively, and are considered positive if they produce tension on the inner side of the shell.

TABLE I

PRESSURE [p.s.i. (Pa x 10^3)] AT SADDLE-SHELL INTERFACE

Welded Case

in (cm) / ϕ DEG	Distance from Saddle Center										
	15 (38.1) Edge 2	12 (30.5)	9 (22.9)	6 (15.2)	3 (7.6)	0 (0) Saddle Center	-3 (-7.6)	-6 (-15.2)	-9 (-22.9)	-12 (-30.5)	-15 (-38.1) Edge 1
0	-50* (-340)	70 (480)	-30 (-210)	0 0	-10 (-70)	0 0	-10 (-70)	-10 (-70)	-50 (-350)	140 (700)	-430 (-2970)
5	-250 (-1730)	80 (550)	-30 (-210)	0 0	-10 (-70)	0 0	-10 (-70)	-10 (-70)	-60 (-420)	180 (1240)	-590 (-4080)
10	-200 (-1380)	60 (420)	30 (210)	0 0	-10 (-70)	0 0	-10 (-70)	-10 (-70)	-50 (-350)	140 (700)	-440 (-3040)
15	-230 (-1590)	70 (480)	-30 (-210)	0 0	-10 (-70)	0 0	-10 (-70)	-10 (-70)	-60 (-420)	170 (1180)	-590 (-4080)
20	-170 (-1180)	60 (420)	-30 (-210)	0 0	-10 (-70)	0 0	-10 (-70)	-10 (-70)	-50 (-350)	140 (650)	-440 (-3040)
25	-200 (-1380)	60 (420)	-30 (-210)	0 0	-10 (-70)	0 0	-10 (-70)	-10 (-70)	-60 (-420)	170 (1180)	-580 (-4010)
30	-140 (-970)	50 (350)	-20 (-140)	0 0	-10 (-70)	0 0	-10 (-70)	-10 (-70)	-50 (-350)	130 (900)	-430 (-2970)
35	-150 (-1040)	50 (350)	-20 (-140)	0 0	-10 (-70)	0 0	-10 (-70)	-10 (-70)	-60 (-420)	150 (1040)	-540 (-3730)
40	-100 (-690)	30 (210)	-20 (-140)	0 0	-10 (-70)	0 0	-10 (-70)	0 0	-40 (-280)	110 (760)	-390 (-2700)
45	-70 (-480)	10 (70)	-20 (-140)	-10 (-69)	-10 (-70)	-10 (-70)	-20 (-140)	0 0	-50 (-350)	110 (760)	-480 (-3320)
50	10 (70)	-20 (-140)	-10 (-70)	-10 (-69)	-10 (-70)	0 0	-10 (-70)	0 0	-30 (-210)	70 (480)	-300 (-2070)
55	70 (480)	-80 (-550)	-10 (-20)	-40 (-279)	-30 (-210)	-40 (-280)	-40 (-280)	-30 (-210)	-50 (-350)	50 (350)	-390 (-2700)
60	360 (2490)	-110 (-760)	70 (480)	20 (138)	30 (210)	-30 (-210)	30 (210)	-30 (-210)	-40 (-280)	60 (420)	-80 (-550)
65	-60 (-410)	-300 (-2070)	-150 (-1040)	-200 (-1380)	-190 (-1310)	-200 (-1380)	-210 (1450)	-200 (-1380)	-270 (-1870)	-90 (-620)	-860 (-5940)
70	2620 (18140)	50 (350)	600 (4150)	380 (2626)	480 (3320)	440 (3040)	500 (3460)	450 (3110)	610 (-4220)	480 (3320)	2230 (-15410)
75	2850 (19690)	1450 (10020)	2010 (13890)	-800 (-5528)	-1280 (8850)	-1090 (-7530)	-1460 (-10090)	-790 (-5460)	-3050 (-21080)	-3000 (-20730)	-5050 (-34900)

*Negative signs indicate compression.

TABLE II

PRESSURE [p.s.i. (Pa x 10^3)] AT SADDLE-SHELL INTERFACE

Unwelded Case

In (cm) / ϕ DEG	Distance from Saddle Center										
	-15 (38.1) Edge 2	-12 (30.5)	-9 (22.9)	-6 (15.2)	-3 (7.6)	0 (0) Saddle Center	-3 (-7.6)	-6 (-15.2)	-9 (-22.9)	-12 (-30.5)	-15 (-38.1) Edge 1
0	-370 (-2560)	0 0	-110 (-760)	-100 (-690)	-130 (-900)	-130 (-900)	-200 (-1380)	-180 (-1240)	0 0	0 0	-2100 (-14510)
5	-440 (-3040)	0 0	-100 (-690)	-100 (-690)	-130 (-900)	-130 (-900)	-200 (-1380)	-180 (-1240)	0 0	0 0	-2020 (-13960)
10	-370 (-2560)	0 0	-110 (-760)	-90 (-620)	-130 (-900)	-120 (-830)	-200 (-1380)	-170 (-1170)	0 0	0 0	-2140 (-14790)
15	-430 (-2970)	0 0	-120 (-830)	-90 (-620)	-130 (-900)	-120 (-830)	-200 (-1380)	-180 (-1240)	0 0	0 0	-1990 (-13750)
20	-320 (-2210)	0 0	-100 (-690)	-90 (-620)	-120 (-830)	-120 (-830)	-200 (-1380)	-170 (-1170)	0 0	0 0	-2070 (-14300)
25	-370 (-2560)	0 0	-100 (-690)	-90 (-620)	-120 (-830)	-120 (-830)	-190 (-1310)	-170 (-1170)	0 0	0 0	-1900 (-13100)
30	-280 (-1930)	0 0	-110 (-760)	-80 (-550)	-120 (-830)	-110 (-760)	-180 (-1240)	-160 (-1100)	0 0	0 0	-1970 (-13610)
35	-310 (-2140)	0 0	-100 (-690)	-80 (-550)	-120 (-830)	-110 (-760)	-170 (-1170)	-160 (-1100)	0 0	0 0	-1780 (-12300)
40	-190 (-1310)	0 0	-80 (-550)	-80 (-550)	-110 (-760)	-110 (-760)	-160 (-1100	-160 (-1100)	0 0	0 0	-1810 (-12500)
45	-210 (-1450)	0 0	-80 (-550)	-90 (-620)	-110 (-760)	-120 (-830)	-160 (-1100)	-170 (-1170)	0 0	0 0	-1610 (-11120)
50	-50 (-350)	0 0	-40 (-280)	-80 (-550)	-80 (-550)	-100 (-690)	-130 (-900)	-160 (-1100)	0 0	0 0	-1580 (-10920)
55	0 0	0 0	-160 (-1110)	-150 (-1040)	-170 (-1180)	-160 (-1100)	-150 (-1040)	-260 (-1800)	0 0	0 0	-1400 (-9680)
60	0 0	0 0	0 0	0 0	0 0	-110 (-760)	-50 (-350)	-190 (-1310)	0 0	0 0	-1610 (-11130)
65	0 0	0 0	0 0	0 0	0 0	0 0	0 0	0 0	0 0	0 0	0 0
70	0 0	0 0	0 0	0 0	0 0	0 0	0 0	0 0	0 0	0 0	0 0
75	-2010 (-13890)	0 0	-10 (-70)	-1140 (-7880)	-640 (-4420)	-890 (-6150)	-800 (-5530)	-1220 (-8430)	0 0	0 0	-3280 (-22670)

*Negative signs indicate compression.

Table III

CIRCUMFERENTIAL PRESSURE DISTRIBUTION
USED BY REFERENCES [4] AND [5]

ϕ DEG	Unwelded	Welded
	p.s.i. (Pa x 10^3)	
0	-200* (-1380)	-200 (-1380)
10	-200 (-1380)	-200 (-1380)
20	-200 (-1380)	-200 (-1380)
30	-200 (-1380)	-200 (-1380)
40	-300 (-2070)	-180 (-1240)
50	-300 (-2070)	-220 (-1520)
60	0 0	0 0
65	0 0	1100 (7600)
75	-860 (-5940)	-1400 (-9670)

*Negative sign indicates compression.

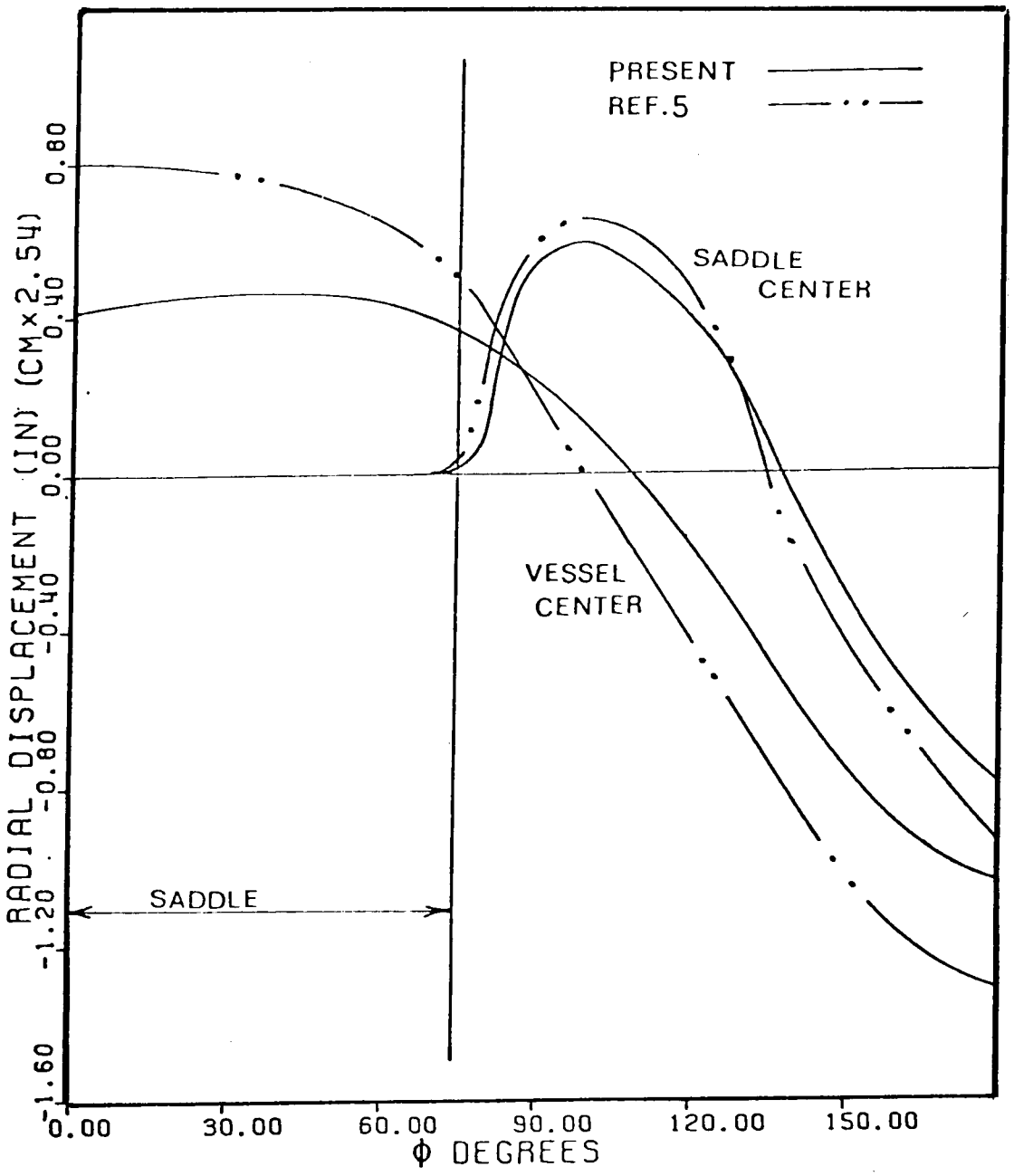

Figure 3. Radial Displacements - Welded

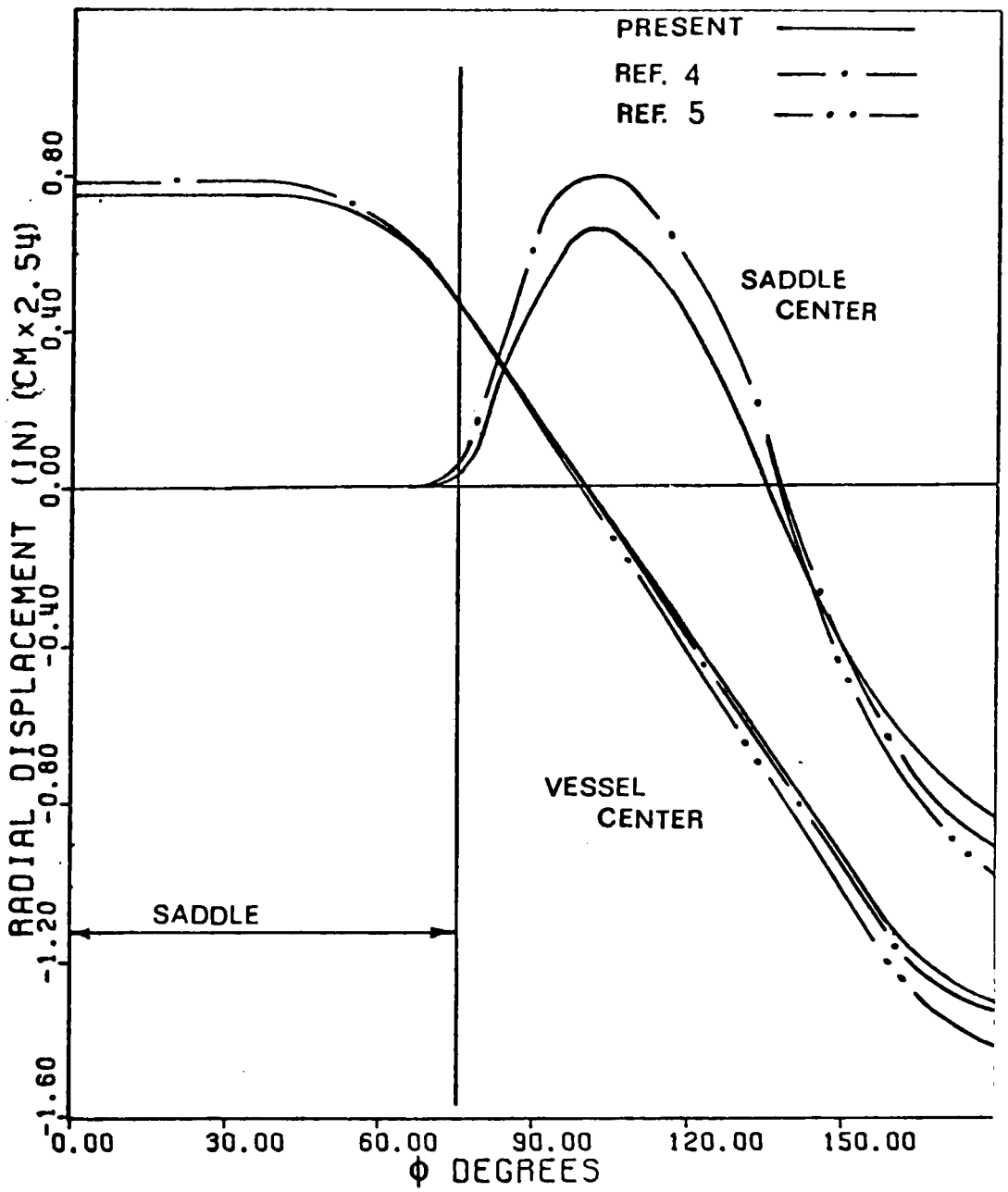

Figure 4. Radial Displacements - Unwelded

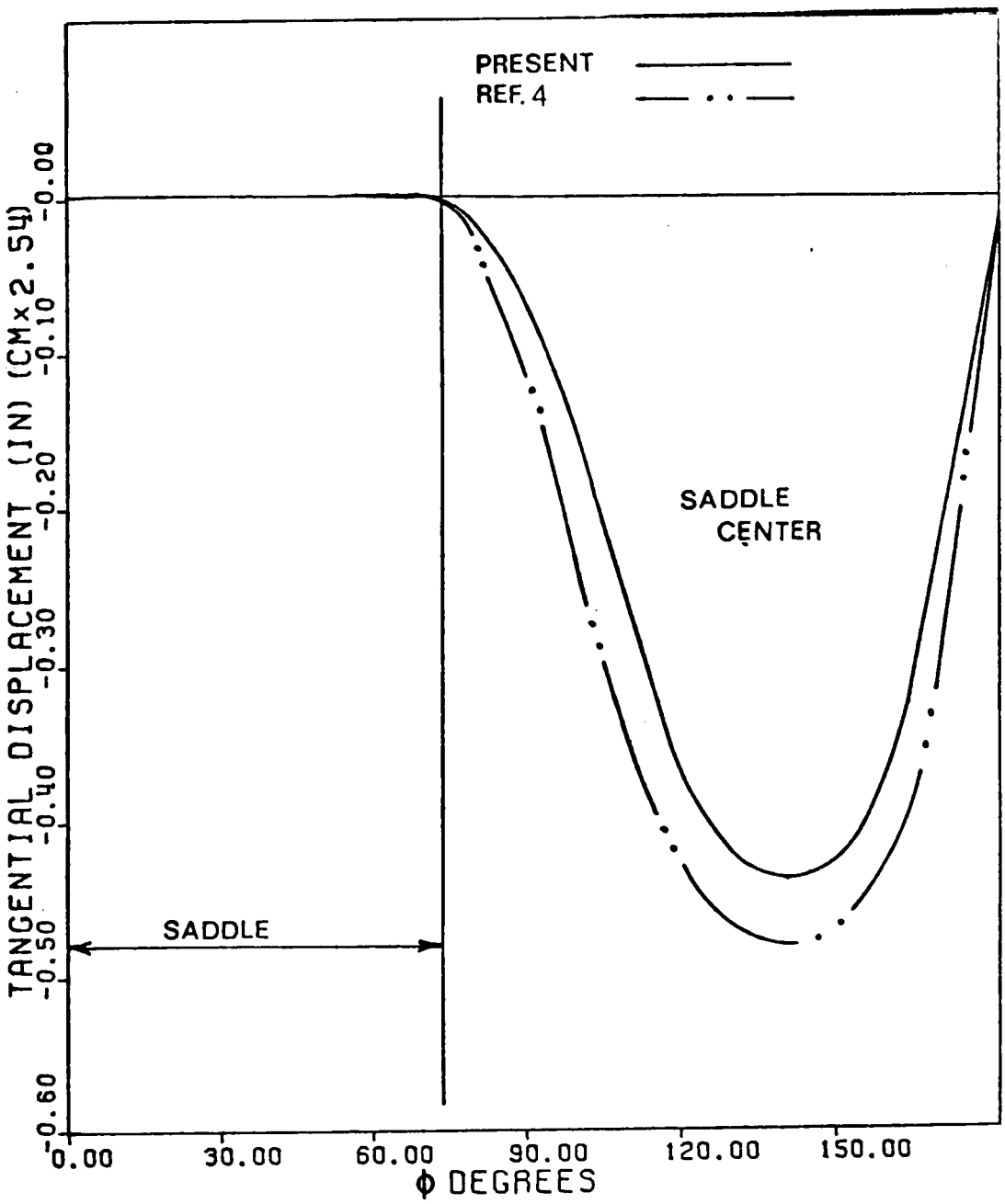

Figure 5. Tangential Displacements - Welded and Unwelded

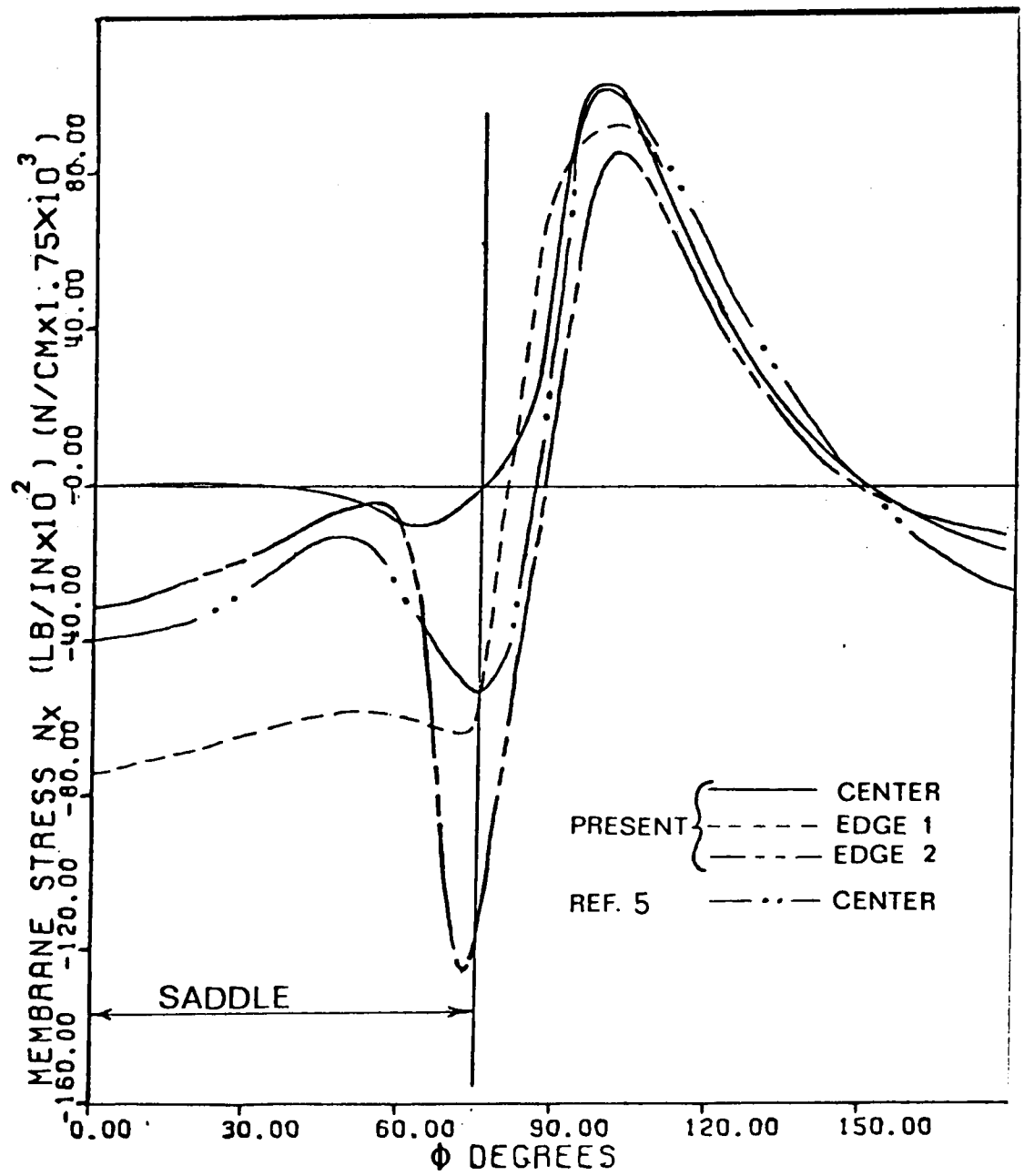

Figure 6. Membrane Stress N_x - Welded

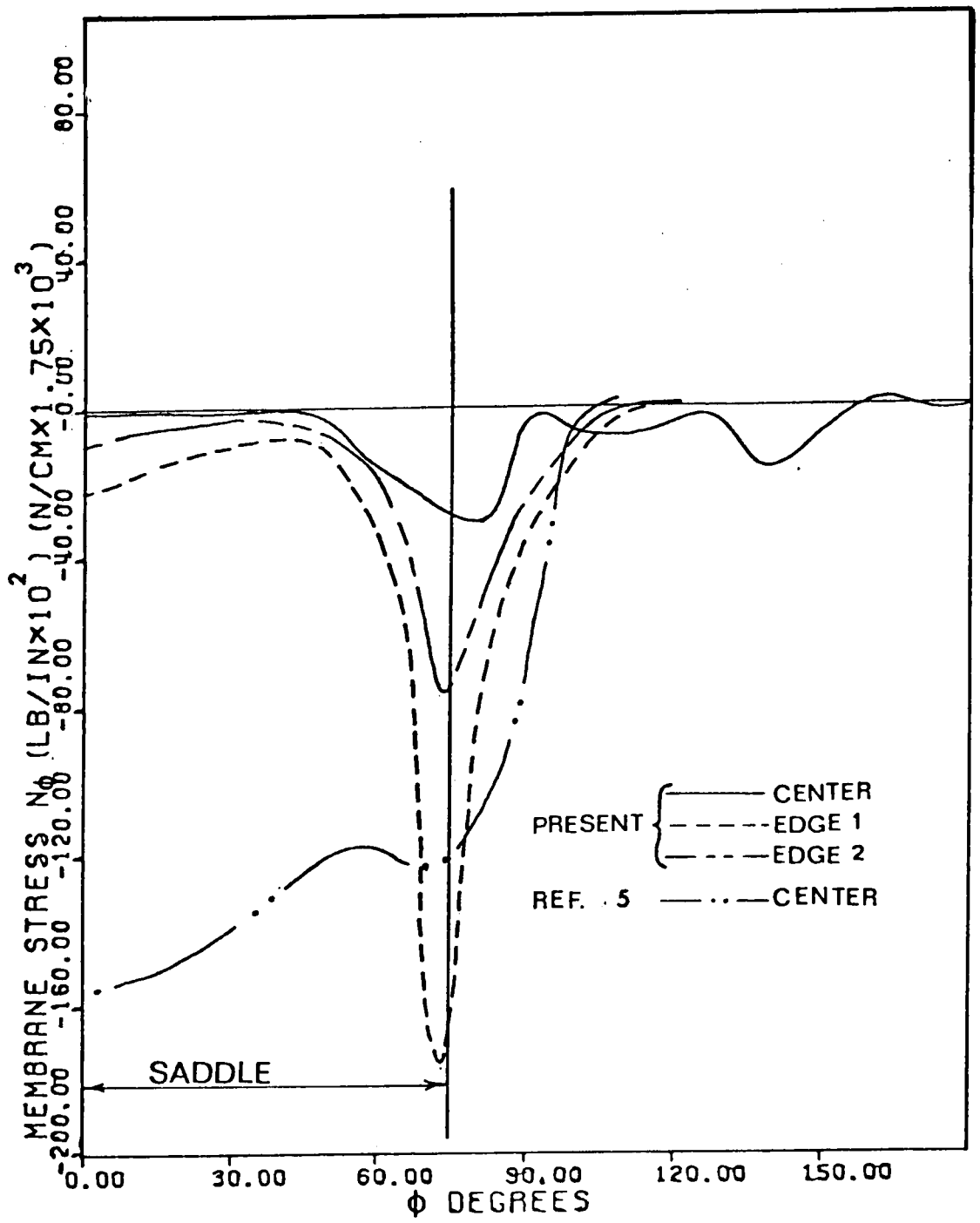

Figure 7. Membrane Stress N_ϕ - Welded

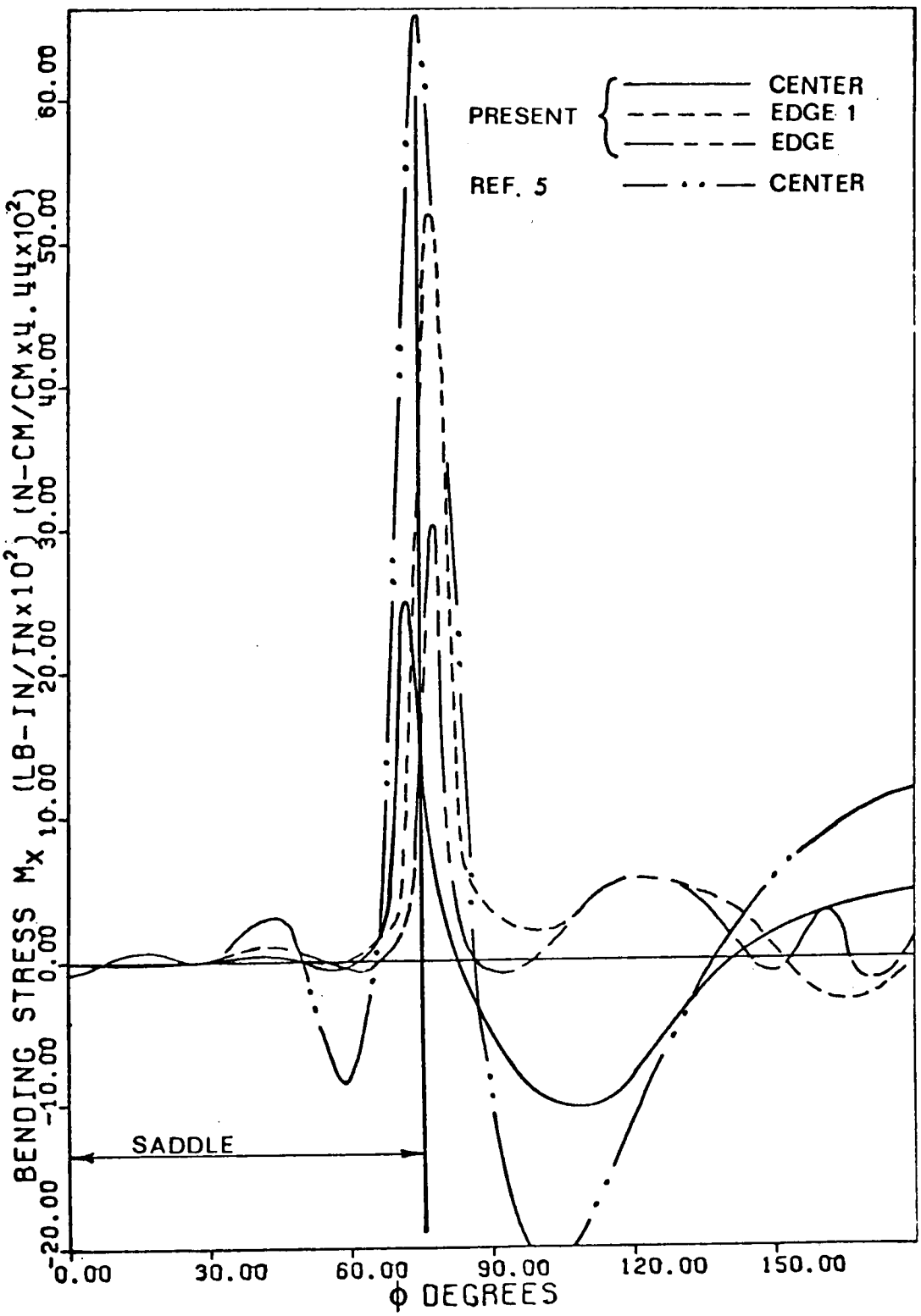

Figure 8. Bending Stress M_x - Welded

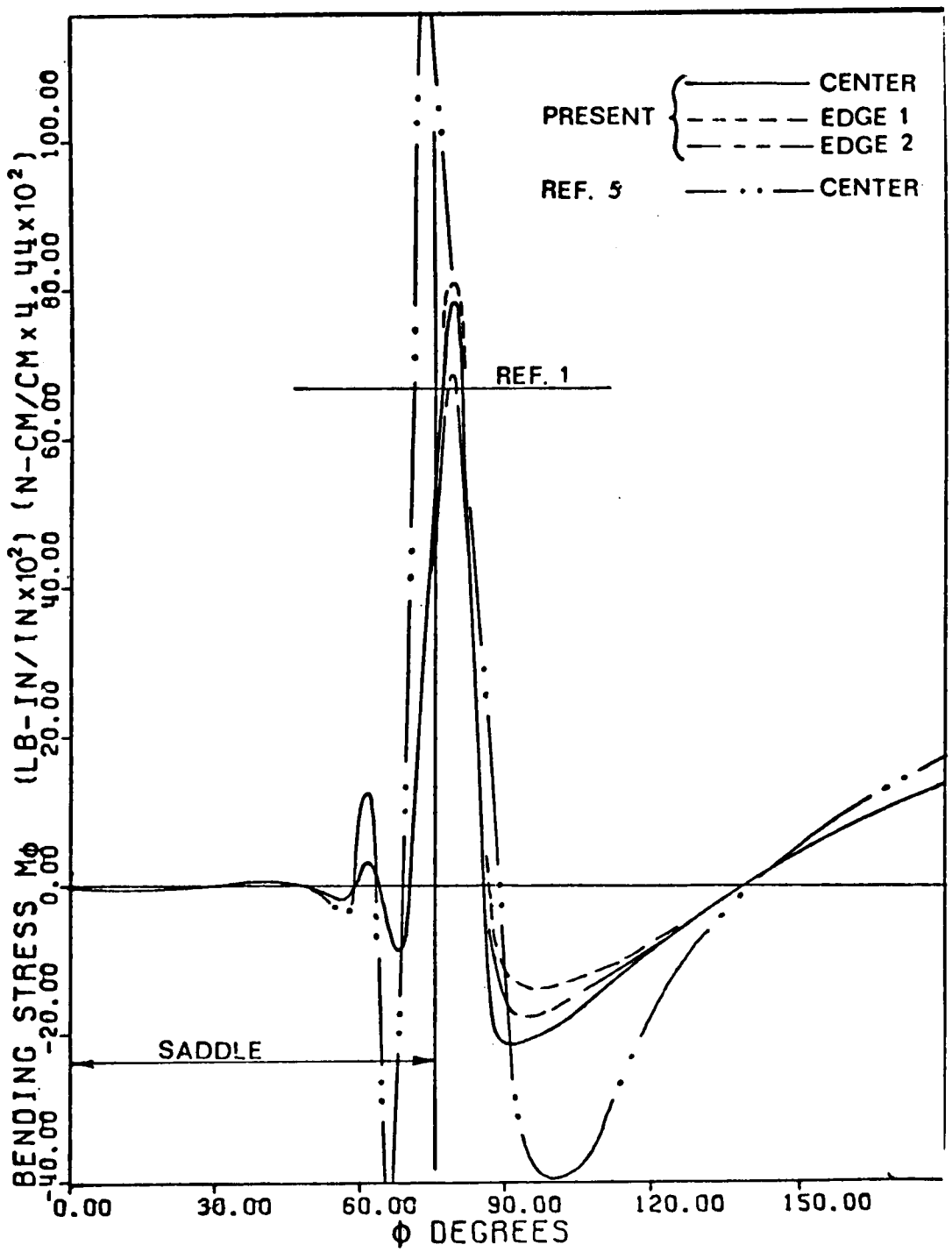

Figure 9. Bending Stress M_{ϕ} - Welded

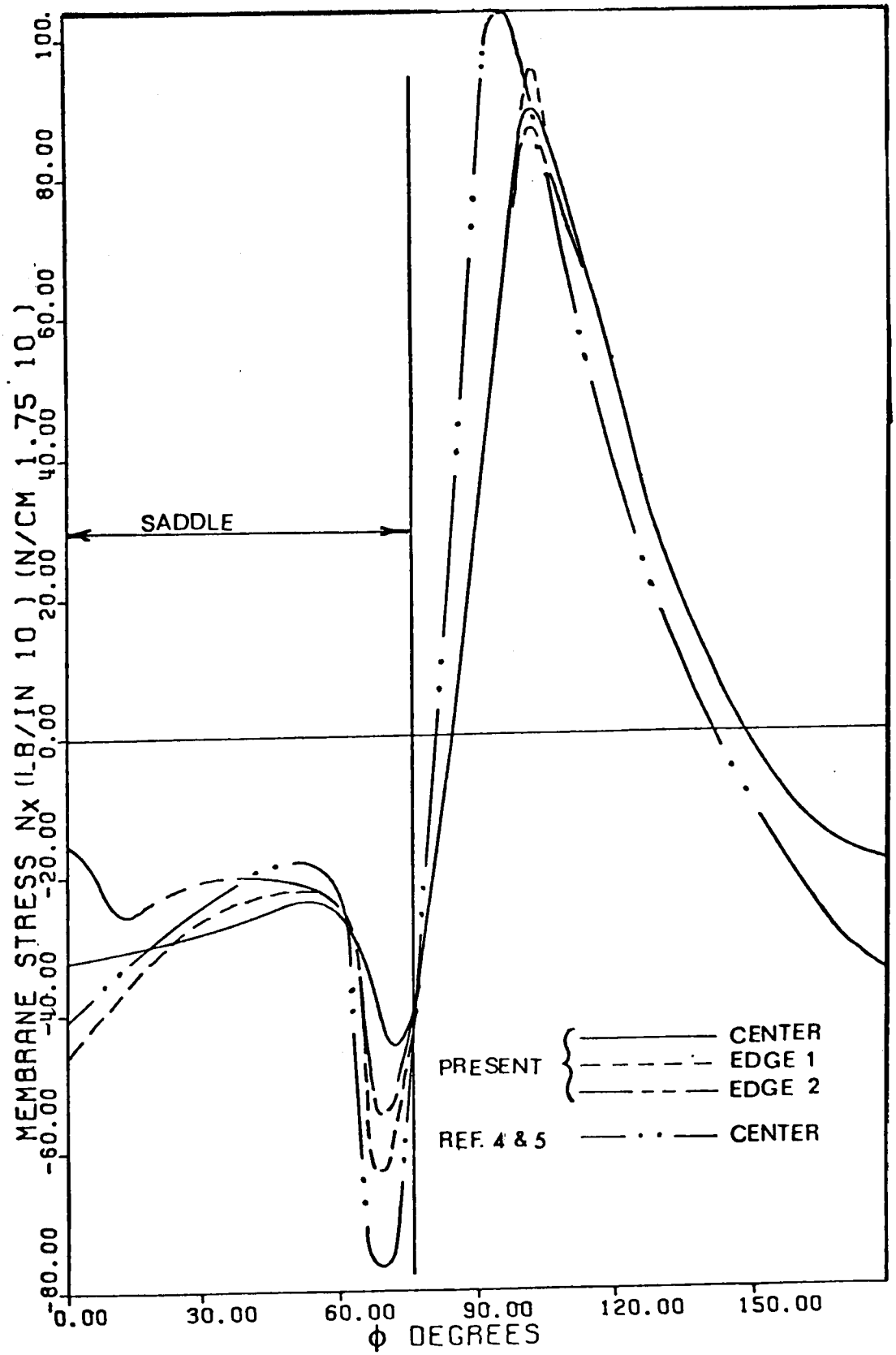

Figure 10. Membrane Stress N_x - Unwelded

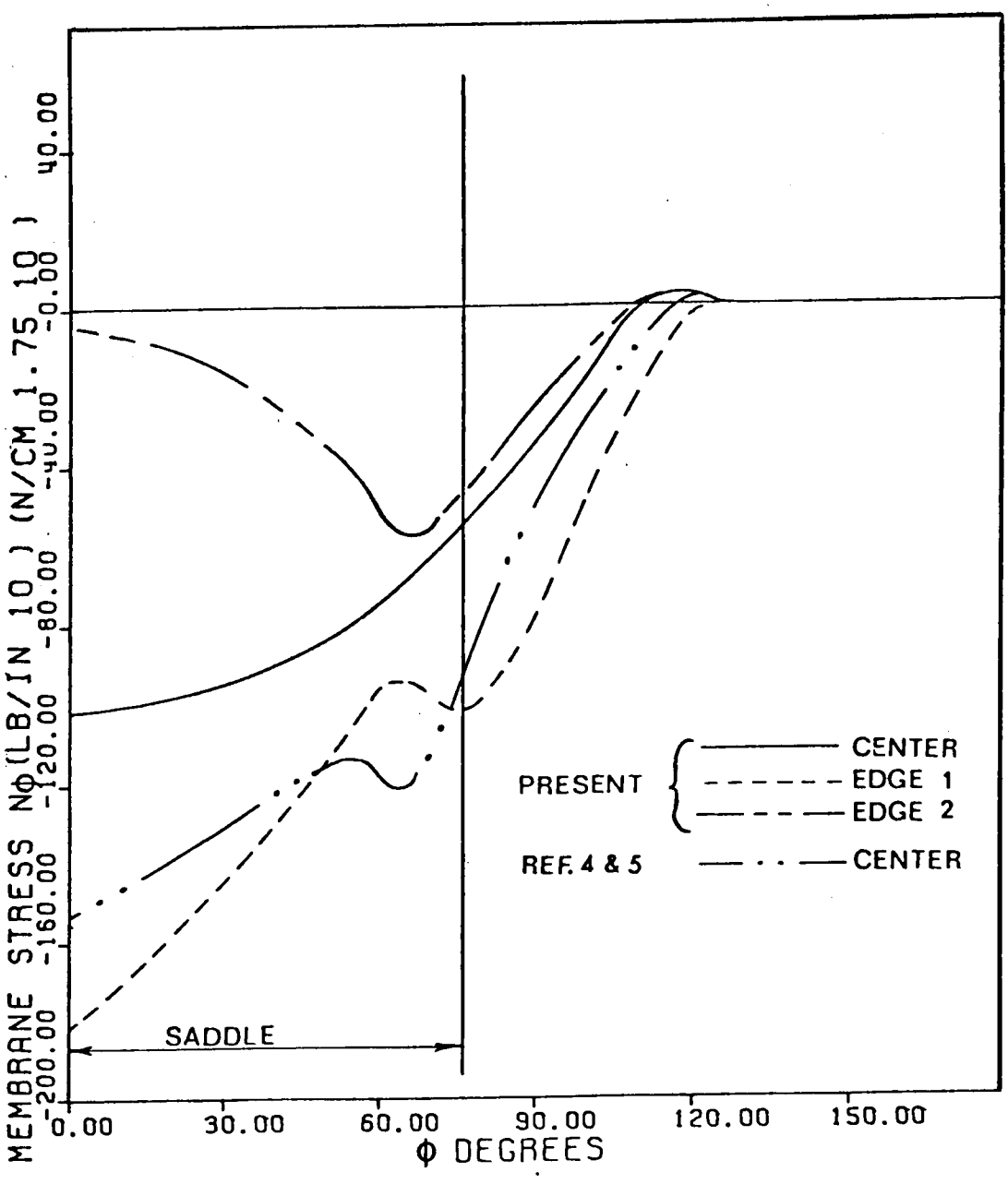

Figure 11. Membrane Stress N_ϕ - Unwelded

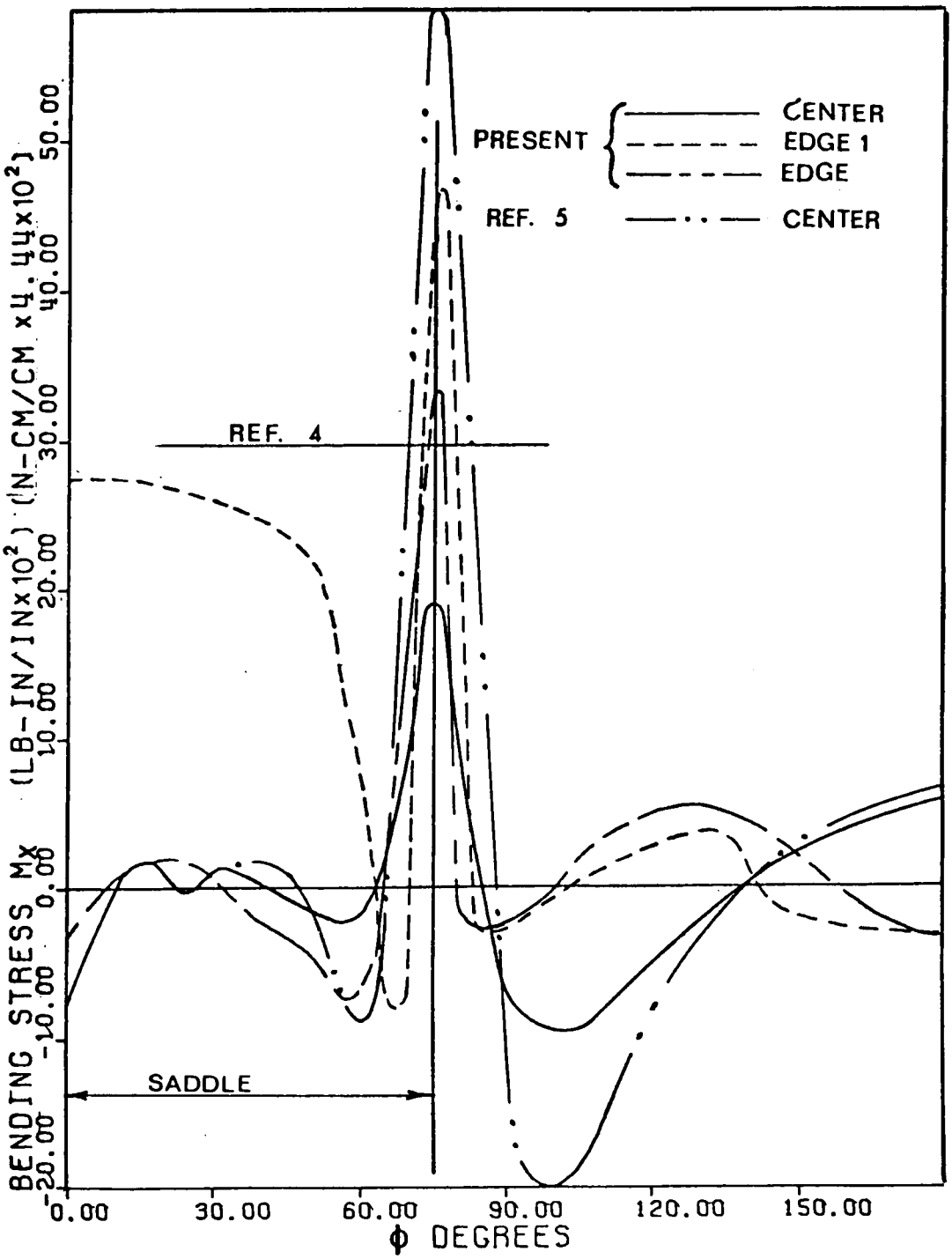

Figure 12. Bending Stress M_x - Unwelded

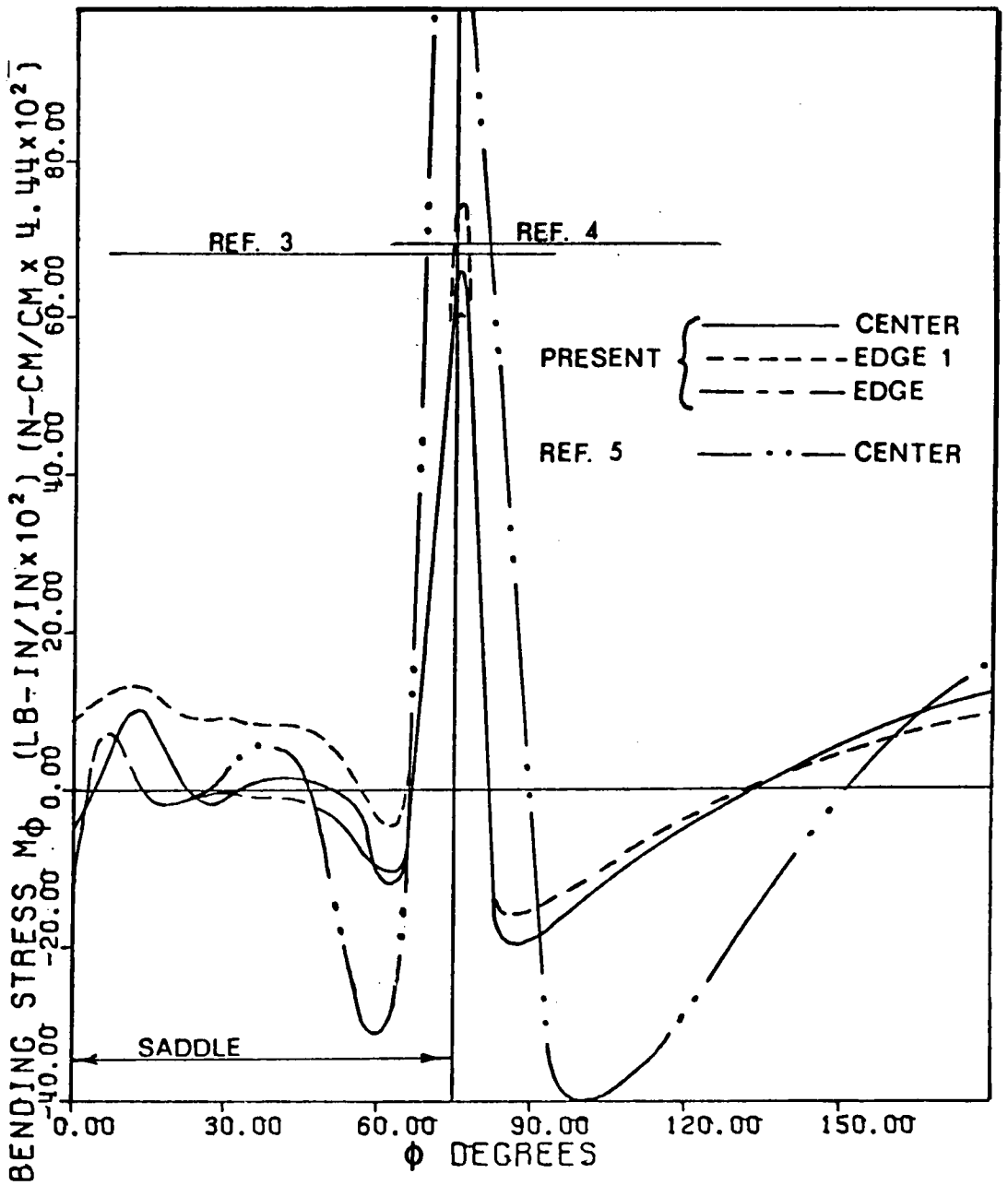

Figure 13. Bending Stress M_{ϕ} - Unwelded

Figures 6 and 7 (see pages 17 and 18) show the membrane stress resultants N_x and N_ϕ for the welded case. There is good agreement with reference (5) except in the saddle region (ϕ = 0 to 75). The present author assumed that the longitudinal and circumferential displacements are restrained around the perimeter of the saddle. The results shown are seemingly consistent with these assumptions. Figure 7 (see page 18) shows that reference (5) obtained a relatively large stress N_ϕ in the saddle region; yet Figure 5 (see page 16) shows that the tangential displacement has been restrained in this region from ϕ = 0 to ϕ = 75. Assuming that the tangential displacement has been restrained along the entire saddle perimeter for this region, it does not seem logical that the vessel should sustain such a large stress at the Saddle Center at ϕ = 0 since the rigid saddle support would resist most of the load.

Figures 8 and 9 (see pages 19 and 20) show the bending stress resultants for the welded case. The maximum longitudinal stress resultant M_x calculated in reference (5) is about 50% higher than that computed by the present author. In addition, the maximum stresses from reference (5) occur at the Saddle Center and those in the present investigation occur along Edge 1. The maximum circumferential bending stress was also computed using Zick's analysis (3). His analysis, which makes no distinction between welded and unwelded supports, predicts a stress about 18% lower than predicted by the finite element analysis.

Figures 10 through 13 (see pages 21 through 24) illustrate the stress resultants for the unwelded case. Figure 10 (see page 21) shows that the longitudinal membrane stress resultant N_x, as computed by the present author, and references (4) and (5) agree closely. Figure 11

(see page 22) shows that the circumferential membrane stress resultant computed by the present author agrees well with references (4) and (5) outside of the saddle region. Inside the saddle region the stress along Edge 1 agrees closely with these other references.

Figures 12 and 13 show the longitudinal and circumferential bending stress resultants, respectively. Since the results from references (4) and (5) were similar except for peak stress values, only peak stress levels are indicated from reference (4). The maximum longitudinal bending stress computed by reference (4) is about 35% less than that computed by the present author. The maximum longitudinal bending stress computed by reference (5) is about 30% higher than that given by the present author. The maximum circumferential bending stresses computed by references (3) and (4) are almost the same and are about 10% less than those given by the present investigation. The maximum circumferential bending stress computed by reference (5) is about 65% higher than that given by the present author.

PART II

STRESS ANALYSIS OF THE SUPPORT DESIGN FOR A HORIZONTAL PRESSURE VESSEL SUBJECTED TO SEISMIC AS WELL AS PRESSURE LOADS

CHAPTER IV

SUPPORT DESIGN FOR CYLINDRICAL VESSELS

I. SADDLE DESIGN CRITERIA

For engineers engaged in the design of saddle supported, horizontal, cylindrical pressure vessels, two regions of critical concern in any stress analysis of the structure are the saddle-shell interface region and the saddle support (see Figure 1, page 7). This method commonly used by design engineers to compute the stress state in these regions is presented by Zick, et al.(1, 2, 3). These analyses are based upon the assumptions that the support in question is rigid and is not fastened to the pressure vessel, and that the design loads consist of pressure loads plus vertical loads, such as dead weight or nozzle loads. Because a large number of pressure vessels constructed today are required to withstand significant seismic loading, the pressure vessel and its support must be designed to resist sizeable horizontal loads as well as vertical loads. Engineers usually attempt to meet these requirements by welding the saddle support to the shell and then securely bolting the support to an appropriate anchoring structure. When the vessel is welded to the saddle support, the support is required to transfer any horizontal load to the support anchors and must, therefore, be designed to sustain this loading. The American Society of Mechanical Engineers (A.S.M.E.) Pressure Vessel Code requires that the support be built according to sound engineering principles; and since pressure vessels are essential to the safety and continuity of operations in such

critical national industries as petroleum processing plants and public utilities, including nuclear power plants, it is certainly imperative that supports be properly designed to ensure that the function of the pressure vessel not be impaired because of a seismic disturbance. The following criteria plays an important role in designing the supports.

Number of Supports

Even though the number of supports generally depends upon the size of the vessel, vessels on a two-saddle support system have certain distinct advantages over a multi-support system. If more than two supports are used the interior supports might resist more of the load than the end supports. This could cause high stresses at the interior support region since these supports are located too far from the head to benefit from its stiffening effect. If only two saddles are used, then one half of the total vertical load will be taken by each saddle and the required analysis of the supporting system is made simpler.

Location of the Supports

The location and type of support depends upon the size of the vessel, the strength of the vessel to be supported, the cost of the support and additional stiffeners if required. Sometimes the location of openings, sumps, etc., in the bottom of the vessel plays an important role in determining the location of the supports. If this is not the situation, then the saddles can be placed according to the designer's requirements. The thin walled vessel with a large diameter which must support its own weight is best supported near the heads to utilize the stiffening effect. It is assumed that if the saddle is located at a

distance of more than half the radius of the vessel measured from the head, the saddle does not benefit from the head. Long, thick walled vessels are not linearly distributed across the thickness of the vessel. Such vessels are best supported where the maximum longitudinal bending stress is nearly equal to the maximum bending stress at the midspan. This is the case assuming the shell is stiff enough to resist bending and is able to transfer the load to the saddle.

Standard Saddle

As shown in Figure 14, standard saddles consist of a web, ribs, wear plate, base plate, and anchor bolts. The center rib, the inner rib, and the outer rib is denoted by CR, IR and OR, respectively.

The basic structure of the saddle which supports the vessel is known as a web. It was found by reference (1) that the web width is not a controlling factor in the vessel design. Hence a nominal width of 12" for steel and 15" for concrete should be adequate. For large, heavy pressure vessels the width should be increased and for small vessels it is desirable to reduce the web width. This is also true for the height of the saddle or the web. The height of the saddle depends upon particular installation requirements. Normally, however, the height of the saddle varies from 6" to 24" depending upon the size and loading conditions of the vessel.

Ribs are provided in the saddle to prevent the saddle from buckling and to provide an additional stiffening effect. The number of ribs depends upon the size of the vessel. However five to seven ribs are generally adequate. In most of the standard saddle designs, one rib is located at the center, two at the extreme ends, and the rest of them in

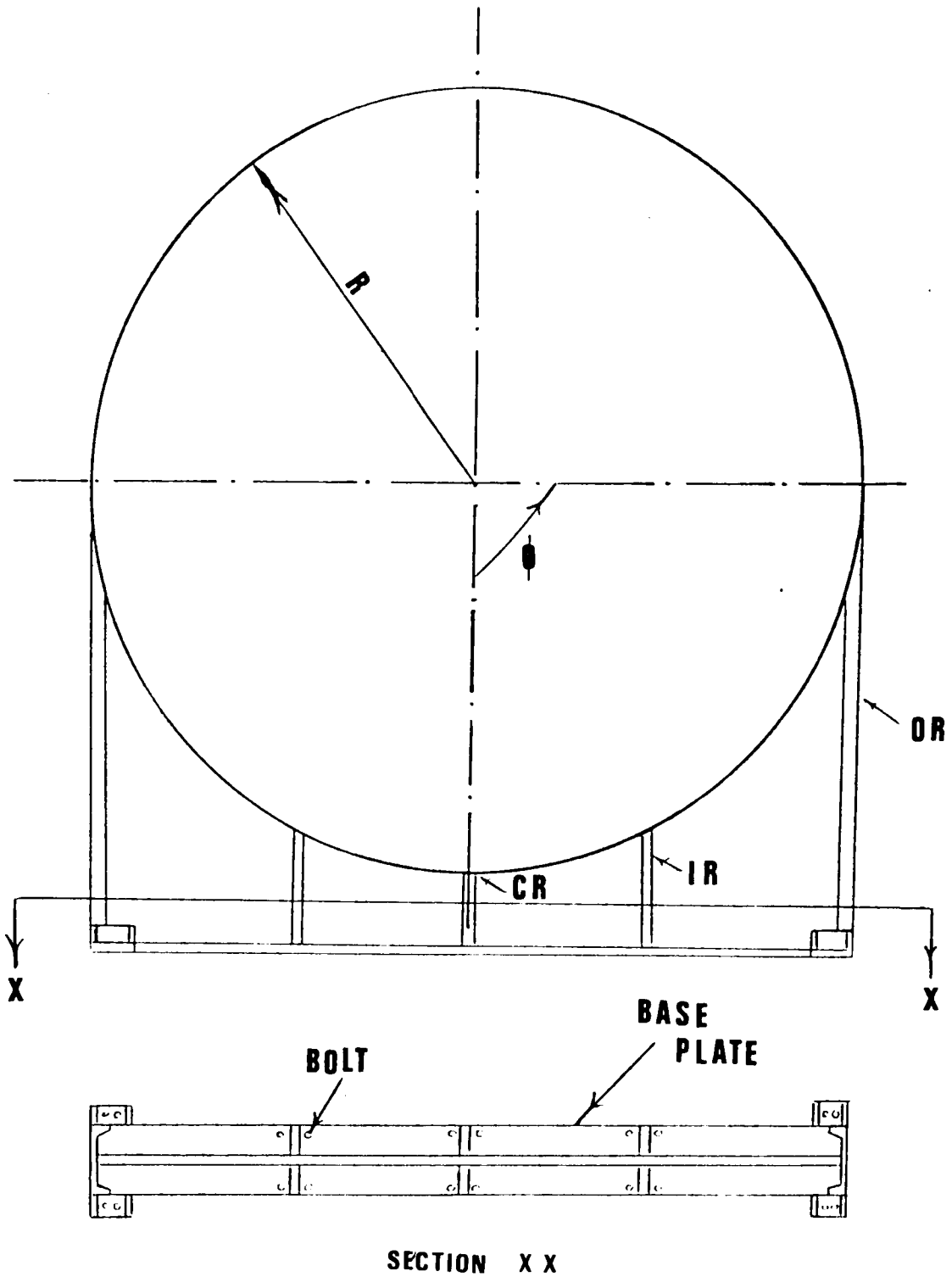

Figure 14. Saddle Design

between the saddle. These ribs are welded to the web and to the base plate at the bottom, and to the wear plate at the top.

At the locations where the thickness of the vessel is uniform and there are no abrupt changes in curvature and geometry, the bending stresses induced are small and can be neglected. The membrane theory has been proved adequate at these locations. However, at the saddle-shell interface the bending stresses are quite high and must not be neglected. This implies that the membrane analysis is not adequate at this location. If these bending stresses exceed the allowable stress, then a larger wear plate and/or ring stiffener should be sprovided at the saddle-shell interface.

The wear plate is generally welded to the cylindrical vessel at the supports. It should be extended above the saddle angle in order to prevent local buckling. The width of the wear plate should be extneded enough so that the shell can resist local deformations. For large, heavy vessels both the wear plates and ring stiffeners should be provided. The ring stiffeners should be designed in such a way as to resist bending to the shell.

The vessel must be fastened to the concrete foundation or to whatever structural frame is used by means of anchor bolts and a base plate. The purpose of anchor bolts is to resist the over-turning moment induced by seismic loading. The bolts should be in multiples of four and should not be spaced too close or else their strength is not fully utilized.

II. INPUT DATA

In order to make the problem realistic the input data for the pressure vessels were obtained from existing designs. These data were also used to perform Zick's analysis (see Appendix) for a comparison with the results obtained in this investigation.

Vessel Data

L	= 240"	tw	= .125"
A	= 12"	ø	= 140 degrees
R	= 57"	Sa	= 12,600 psi
P	= 24 psi	Ds	= .283"
ts	= .25"	Dw	= .03611 lbs/in^3

Saddle Data

b	= 8"	t_1	= .36"
h	= 6"	B	= 83"
n	= 3	k	= 16"
tr	= .38"		
r	= 8"		

III. LOADING CONDITIONS

The vessel was subjected to the following loads: Hydrostatic pressure load; load caused by the weight of the vessel; and horizontal loads.

Hydrostatic Pressure Load

A computer program was developed to generate the hydrostatically varying pressure load in the cylinder. This load was assumed to be positive in the outward normal direction. The computer program determines the hydrostatic pressure load acting at the centroid of the finite element by using the equation:

$$P = (Dw)\ (R)\ (\text{Cos}\ \theta)$$

where θ is an angle between the X-axis and the centroid of the element, R is the radius of the cylinder, Dw is the density of water and P is the pressure.

Weight of the Vessel

The weight of the vessel was introduced in the SAP IV program by specifying the mass density of the material.

Horizontal Loads

The horizontal load which a pressure vessel and its supports must sustain can be vectorially separated into a transverse and longitudinal component. For the type of pressure vessel and support shown in Figure 15, transverse horizontal loads can be assumed to be shared equally by both supports, and one can show with the use of relatively crude engineering assumptions that the supports can adequately resist large transverse seismic loads and were therefore not considered in this investigation. However, the ability of this type of saddle support to resist large longitudinal horizontal seismic loads is questionable. Unlike the transverse load, the longitudinal load must often be resisted by only a

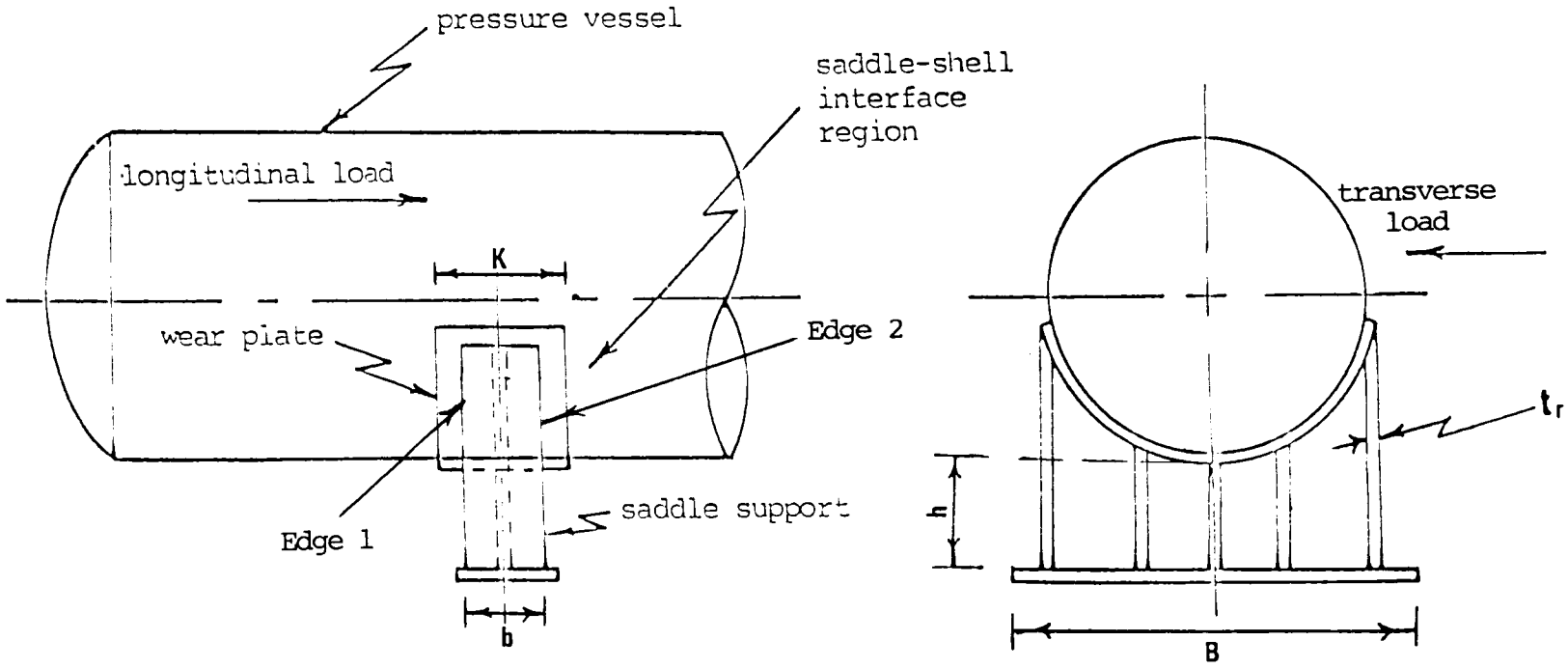

Figure 15. Saddle Support Configuration

single support. This is because thermal expansion considerations may require the anchor bolt holes in all but one of the support base plates to be slotted in the longitudinal direction. A peak acceleration of 0.3 g was assumed in the longitudinal direction. The resulting horizontal load[1] acting in the + Z direction would be:

$$F_h = 0.3\ (Ws + Ww)$$

where F_h is horizontal seismic load

Ws is weight of the shell

Ww is weight of water.

This load was equally distributed along the nodal points of the cylinder at the location where heads would normally be welded to the cylinder.

IV. SOLUTION PROCEDURE

The structural analysis program SAP IV, developed at the University of California, Berkeley, was used for this investigation. Figure 16 describes the geometric configuration of the vessel and its saddle support, and Figure 17 shows the finite element mesh used for this investigation.

Since manual preparation of data for this problem was time consuming and laborious, computer programs were developed to discretize the continuum into quadrilateral and triangular elements. The program was also capable of numbering the nodes and elements and computing the nodel coordinates. As shown in Figure 17, the grids near the saddle-shell interface region were made finer to give an accurate assessment of the

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[1]The pressure vessel and contained fluid are assumed to act as a rigid body.

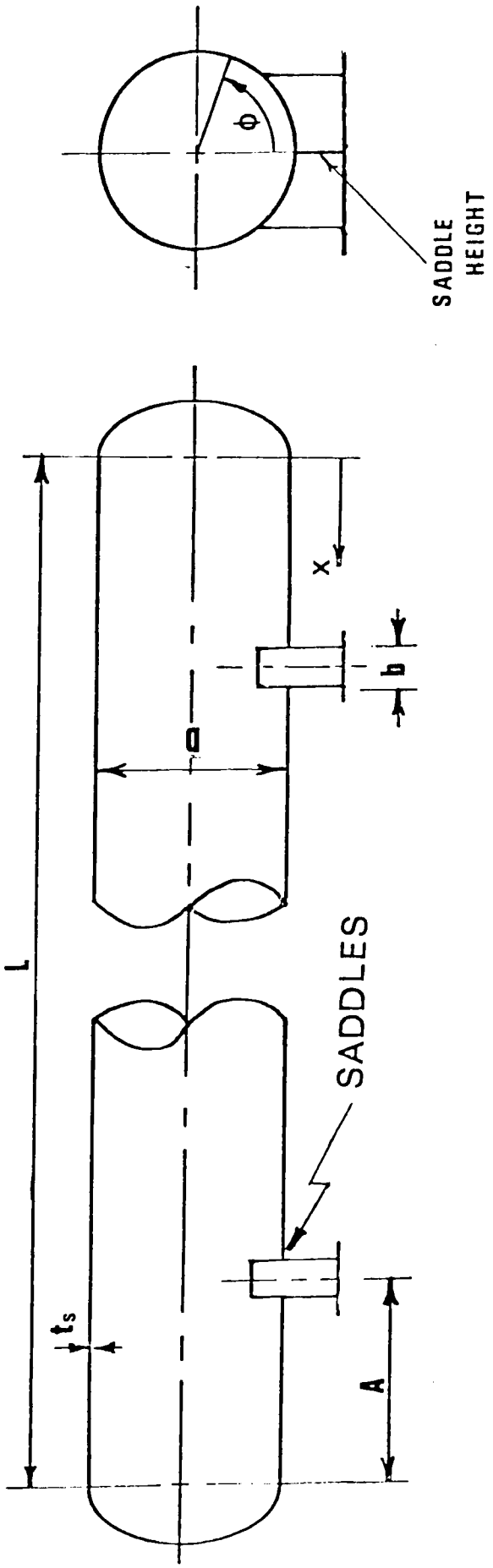

Figure 16. Vessel Dimensions

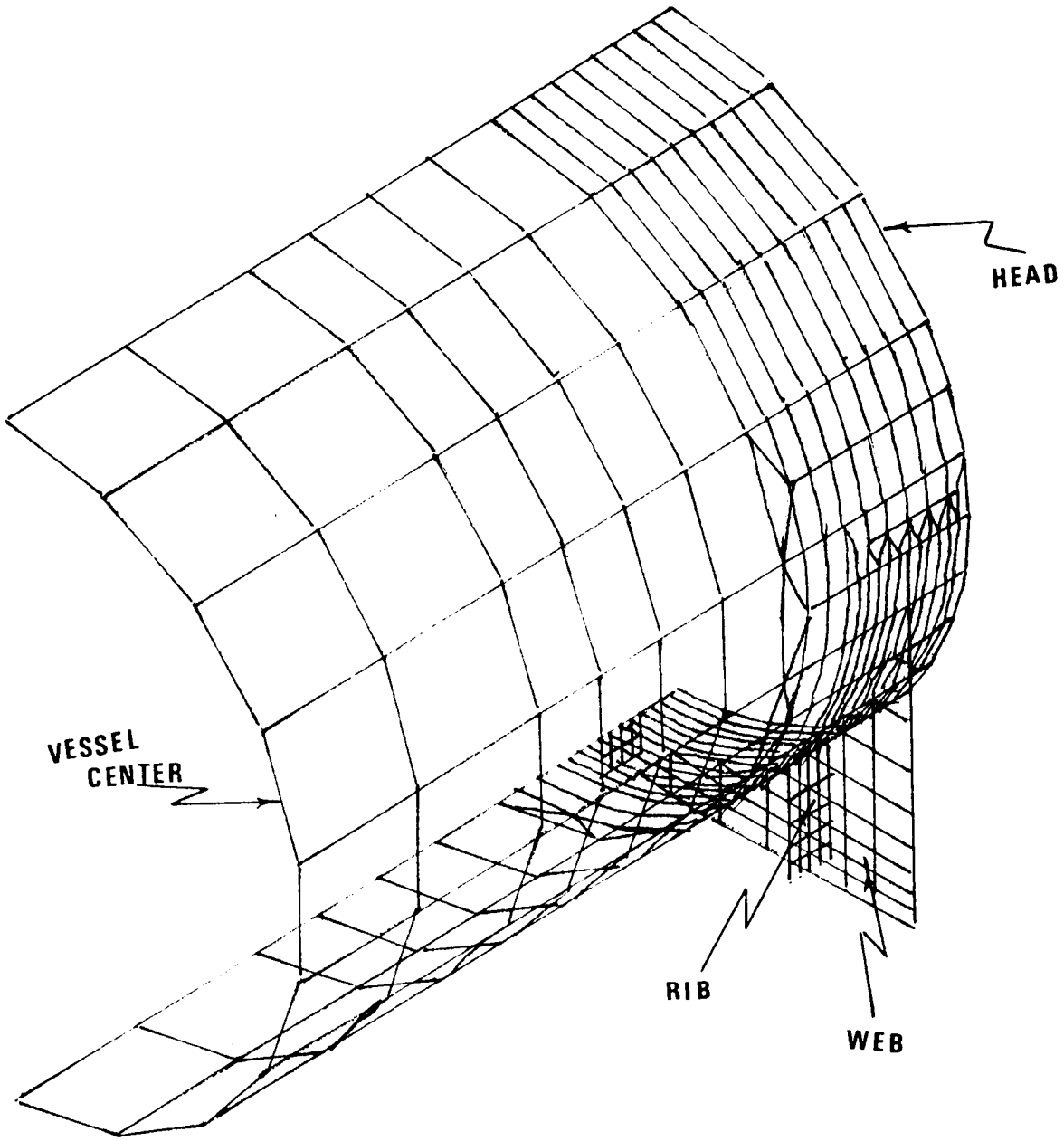

Figure 17. Finite Element Model

stresses in this vicinity. Grids in regions other than the saddle-shell interface area were made gradually coarser by increasing the size of the elements. This fine-to-coarse grid dividing method also helped in minimizing the band width. The fine-to-coarse elements were connected by three node triangular transient elements. However, attempts were made to minimize the use of these triangular elements since the computer program SAP IV assumes linear distribution of displacements for constant strain triangular elements. The thin, flat shell element used in the program was a quadrilateral of arbitrary geometry formed by four compatible triangles. The stresses and moments were calculated at the centroid of the element. The elements had six interior degrees of freedom which were eliminated at the element level, shown in Figure 15, prior to assembly. The resulting quadrilateral had six degrees of freedom per node.

In order to take advantage of the symmetry only one quarter of the vessel was modeled as shown in Figure 17. Due to symmetry, all the nodal points which lie on the X-axis at $\emptyset$ = 0 degrees and $\emptyset$ = 180 degrees were prevented from rotating in the Z-direction and displacing in the Y-direction. In the analysis of the flat plates the stiffness associated with the rotational normal to the surface is not defined in the program. Therefore, whenever two or more elements shake a common node in the same plane, their rotations in the normal direction were restricted. The appropriate boundary conditions at the center of the vessel for horizontal and pressure load cases were applied. The displacement in the radial direction was assumed to be zero for the horizontal load case and the displacement in the longitudinal direction was assumed to be zero for pressure load case.

The effect of the heads on the stiffness of the vessel were obtained by using beam elements with small tortional stiffness and large axial and bending stiffnesses connecting the node points along the circumference of the cylinder at the location where the heads are normally welded to the vessel. The axial force in the cylinder caused by pressure acting on the heads, and the horizontal loads were simulated by applying concentrated loads at these nodes.

Since the ratio of span to depth for the outer rib was quite large, the outer rib was replaced by beam elements of the same stiffness. This change helped in reducing the band-width drastically and eventually the computational time.

The wear plate was assumed to be welded to the cylindrical vessel along its lower periphery. It was modeled by simply adding its thickness to the thickness of the respective shell elements such that it would have the same deformation as those of the respective shell elements.

The use of ring stiffeners at the saddle-shell interface would resist bending moments at this location. This implies that there will not be any bending stresses at the saddle-shell interface as predicted by reference (3). The reason that ring stiffeners were not used in this investigation was to find the true state of stress at the saddle-shell interface.

The base plate was simulated by using beam elements at the bottom of the web. These beam elements were allowed to rotate in the longitudinal direction only. The anchor bolt was replaced by a boundary element which was essentially a spring having an axial displacement stiffener and axial rotational stiffness. These boundary elements were made perpen-

dicular and parallel to the saddle in order to determine the tension and shear load in the anchor-bolt.

Several typical vessels were considered for this investigation. All the parameters except the saddle angle and the height of the saddle were kept constant in all the computer runs. The variation in the parameters was as follows:

i) saddle angle
   a) 180 degrees saddle
   b) 140 degrees saddle

ii) height of the saddle
   a) 6"
   b) 12"
   c) 24"

Six runs were made by varying one parameter and keeping the rest of the parameters constant. Since the vessel had different boundary conditions for horizontal and pressure loads, each computer program was run separately to account for these load cases. Thus twelve computer runs were made for each vessel.

V. RESULTS AND DISCUSSION

For future reference, the circumferential lines on the vessel which pass through the edges of the saddle toward the center of the vessel and toward the head of the vessel are denoted as Edge 1 and Edge 2 respectively. The circumferential line passing through the center of the saddle is denoted as the Saddle Center. The stresses, displacements,

and anchor bolt forces induced by pressure loads and horizontal loads are superimposed in all the figures.

The bending stresses are calculated using the equation

$$Sb = 6 \times M / t^2$$

where M is any of the three bending moments (M_{xx}, M_{yy}, or M_{xy}) in $\frac{lb\text{-}in}{in}$ and t is the thickness of the shell in inches. They are considered positive if they produce tension on the inner side of the shell. These stresses are superimposition of the stresses calculated from pressure loads, dead load of the vessel and swismic loads.

Longitudinal Bending Stresses

Figure 18 shows that the longitudinal bending stresses are proportional to the height of the saddle. Figure 19 represents longitudinal bending stresses S_{bx} for 140 degrees saddle angle and for the saddle heights 6, 12, and 24 inches. These stresses seem to be independent of the saddle height. Regardless of the saddle height and saddle angle, the over-all shape of the curves remains the same in Figures 18 and 19.

The values computed by present study are 400% and 500% higher than those given by reference (3) in Figures 18 and 19, respectively. The discrepancy can be attributed to the fact that the analysis presented by reference (3) is based upon the assumption that the simple beam theory is adequate in determining the longitudinal stresses in the shell. However, it has been recognized that longitudinal bending stresses calculated from simple beam analysis yield poor results which is confirmed by this investigation.

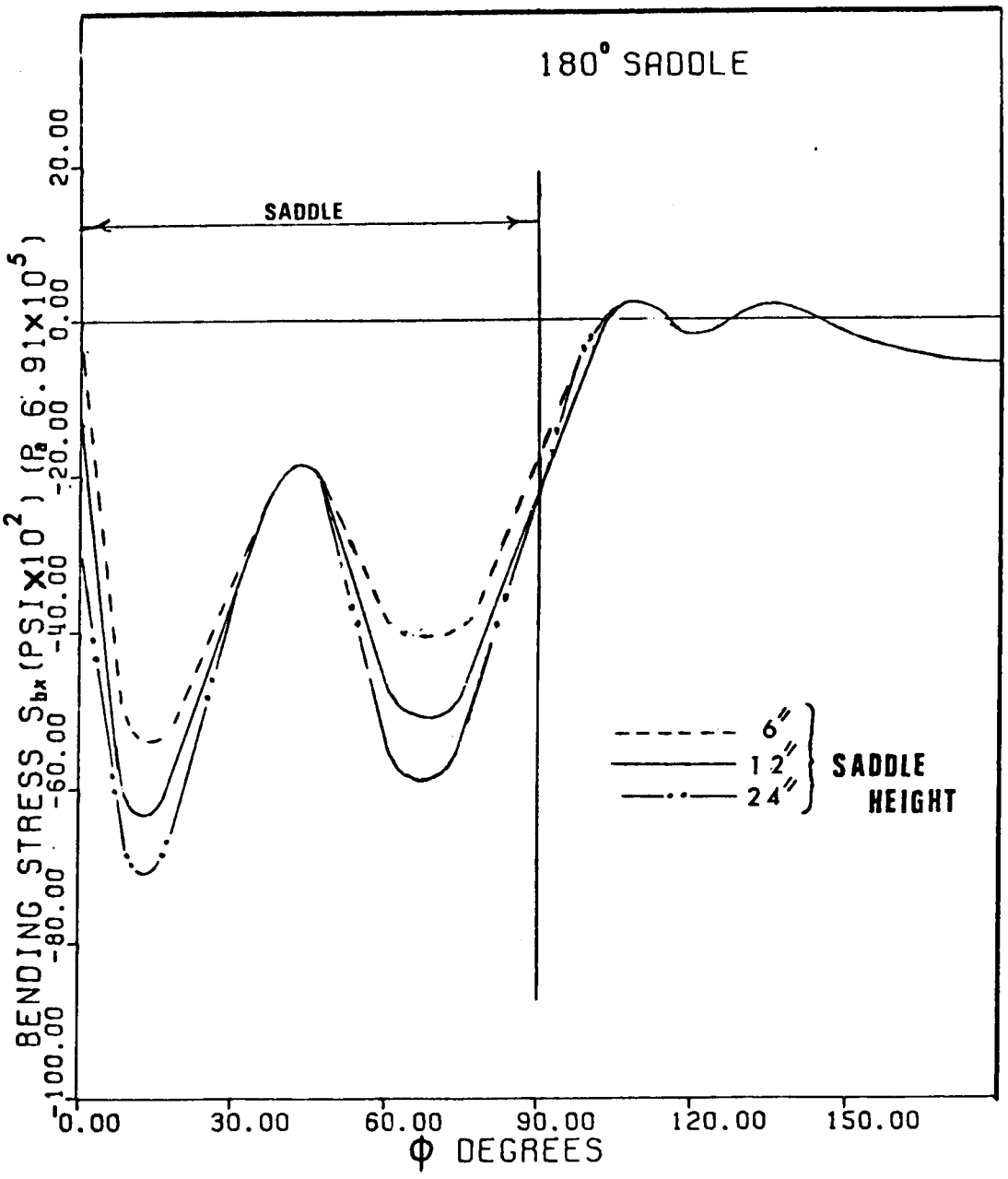

Figure 18. Bending Stress S_{bx} - 180° Saddle Center

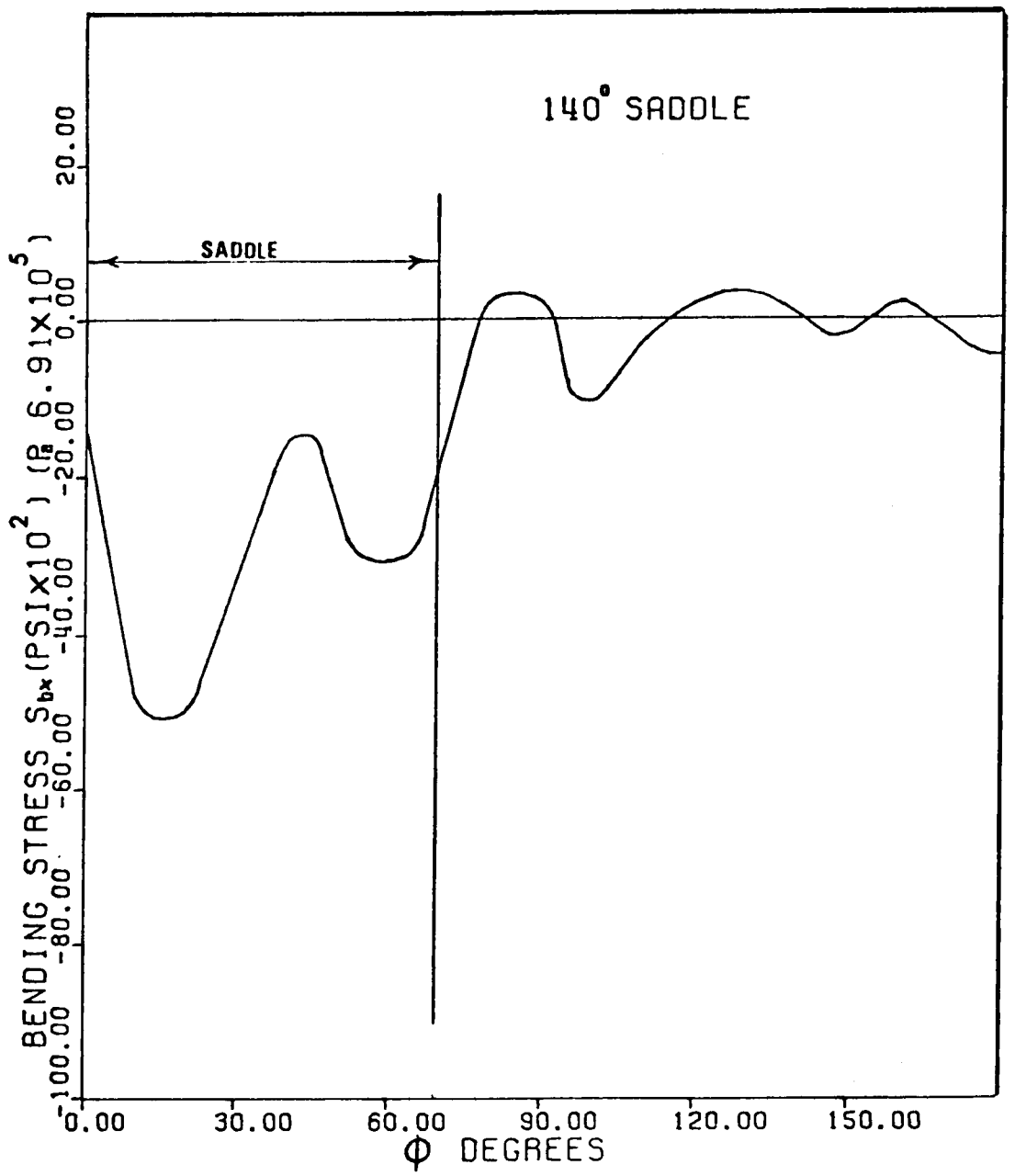

Figure 19. Bending Stress S_{bx} - 140° Saddle Center

The maximum longitudinal stresses are compressive for all cases and occur at the bottom of the shell when ϕ = 20 degrees. The next highest stresses for both the saddle angles occur just below the horn of the saddle. This can be explained by considering that the concentrated load induced by the saddle causes large deformations at the saddle-shell interface. These stresses reduce drastically as ϕ increases above the horn of the saddle.

Circumferential Bending Stresses

As shown in Figures 20 and 21, the maximum circumferential stresses occur just below the saddle horn. The maximum bending stresses calculated using Zick's analysis (3) are 47% and 67% higher than those presented in Figures 20 and 21 respectively.

The inconsistency in the results could be due to the assumption made by reference (3) that the saddle support is rigid and not fastened to the vessel and, moreover, his analysis does not concern itself with seismic loads. Figure 20 shows increases in compressive stresses as a function of saddle height. However, Figure 21 shows increases in tensile stresses as a function of saddle height. This phenomenon is not clearly understood.

Membrane Stresses

Figures 22 and 23 represent membrane stresses S_{mx} at saddle center for saddle heights of 6, 12, and 24 inches, and for saddle angles of 180 degrees and 140 degrees. The change in membrane stresses as a function of saddle height is very small and hence neglected.

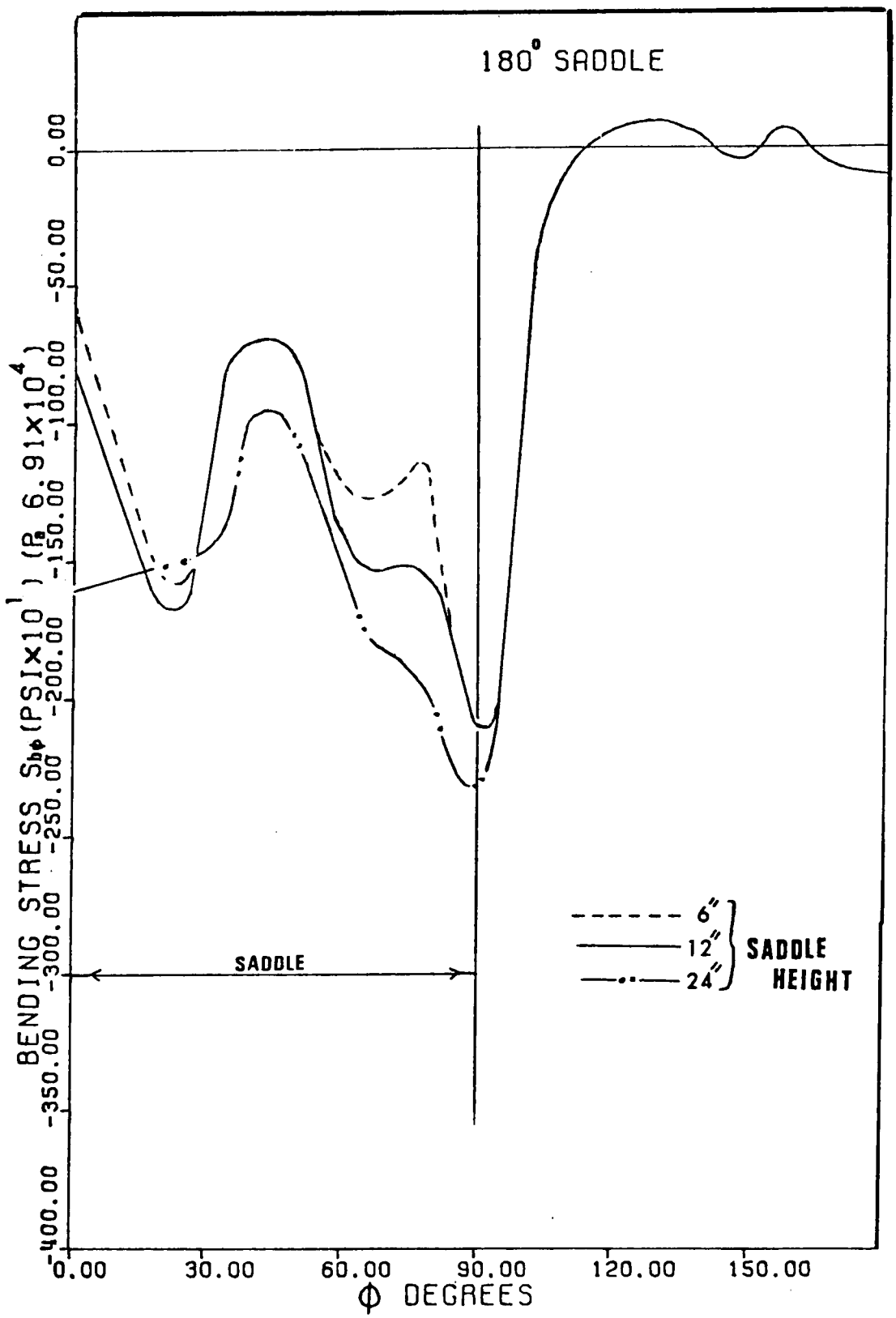

Figure 20. Bending Stress $S_{b\phi}$ - 180° Saddle Center

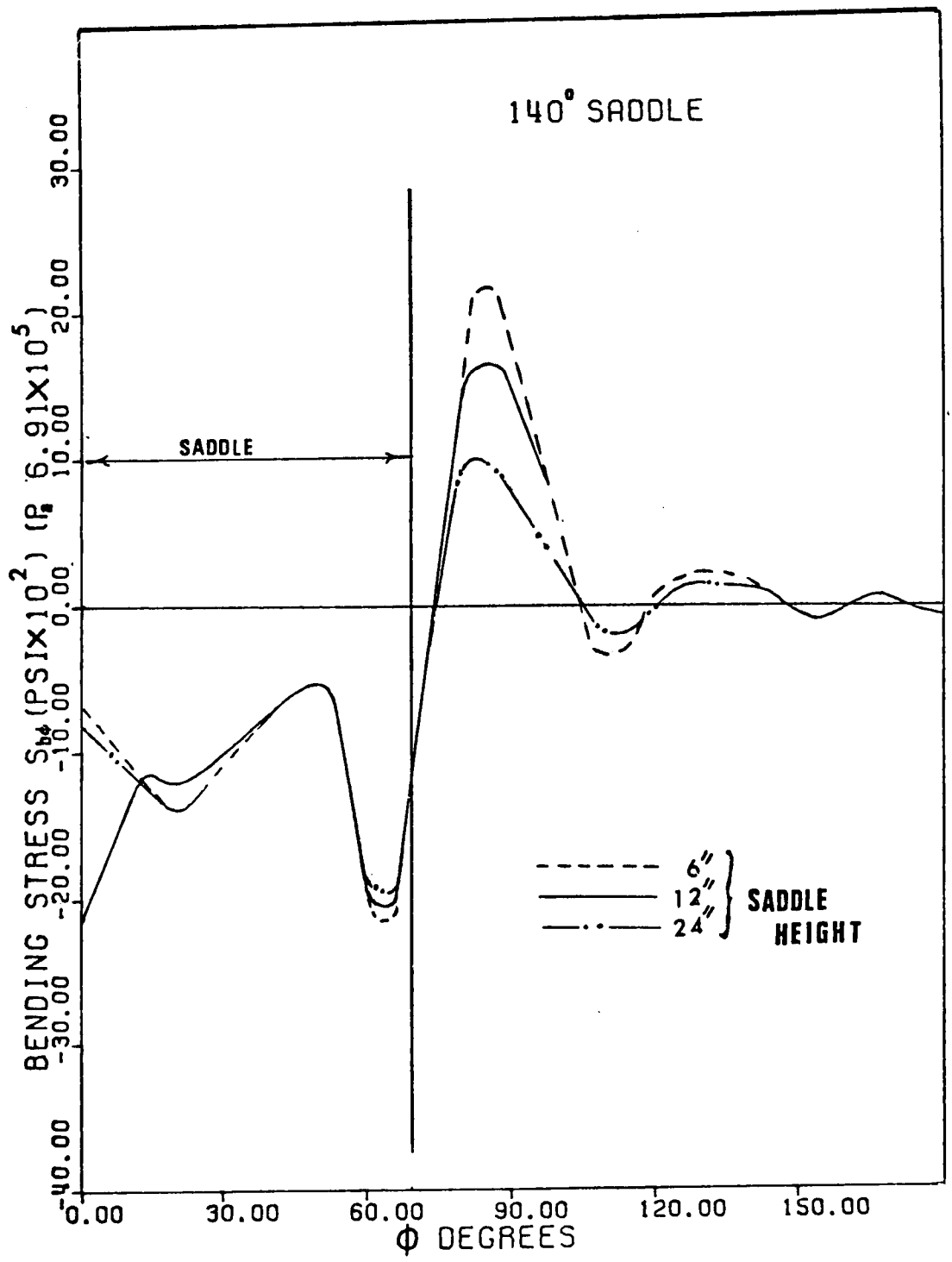

Figure 21. Bending Stress $S_{b\phi}$ - 140° Saddle Center

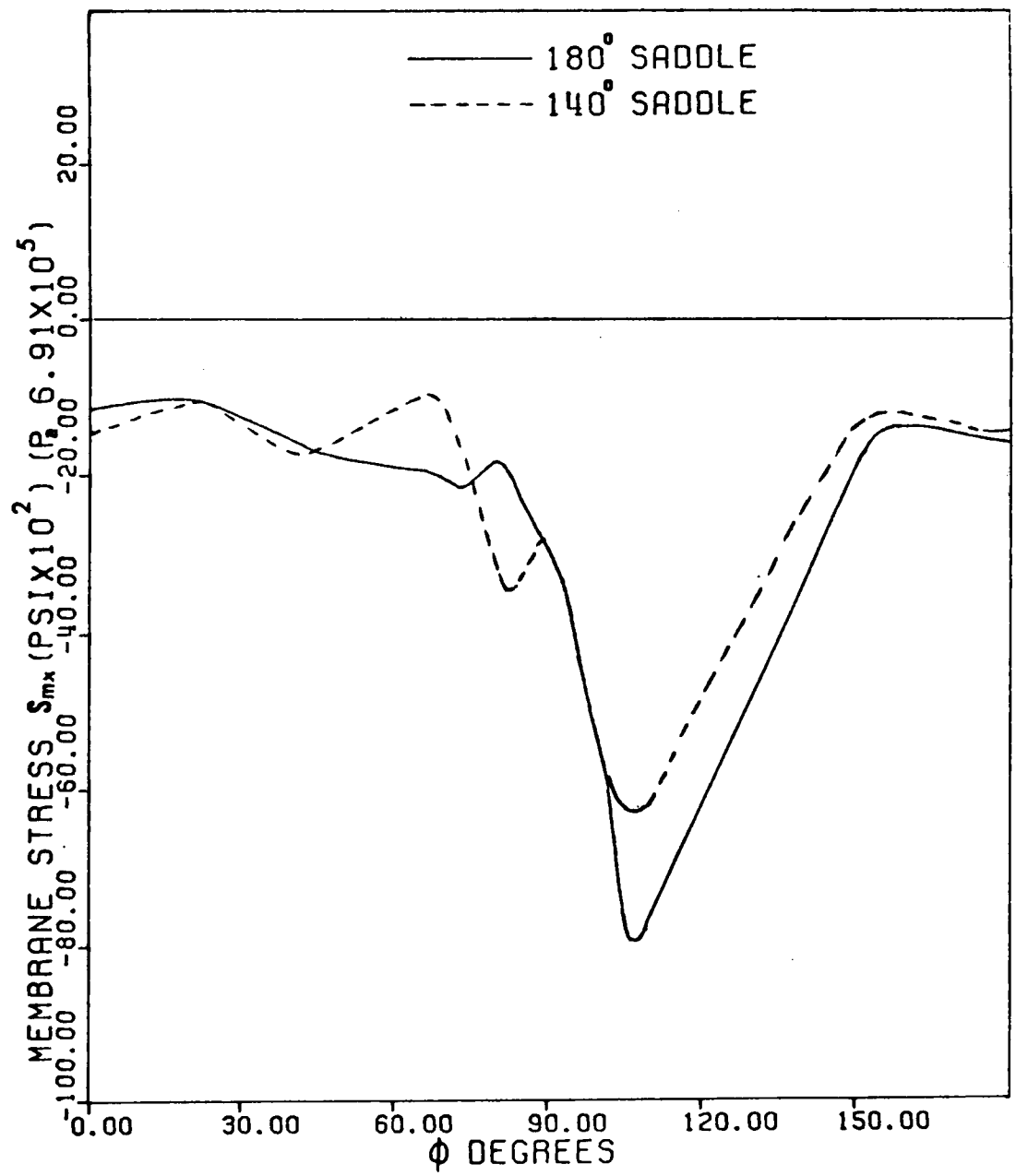

Figure 22. Membrane Stress S_{mx} - Saddle Center

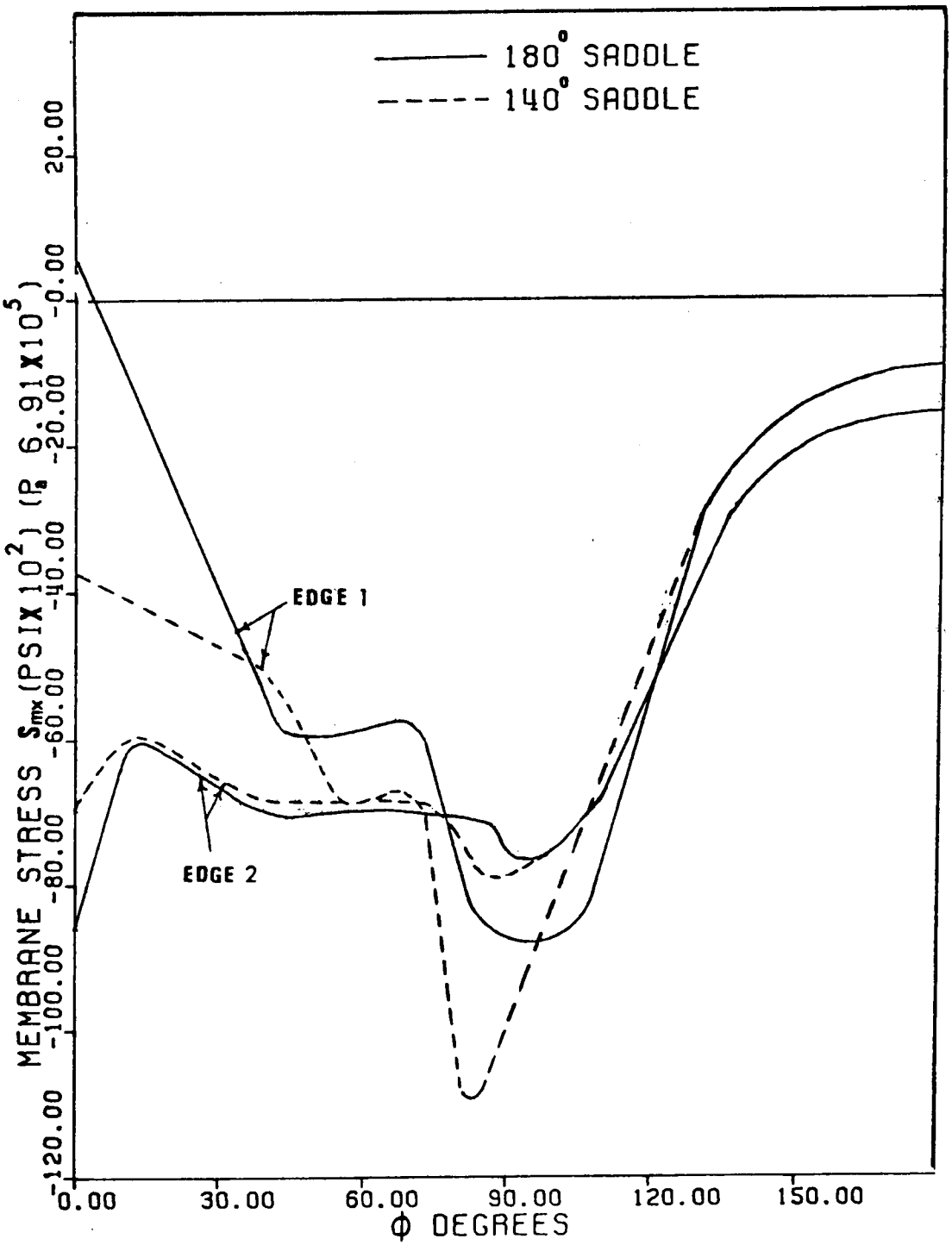

Figure 23. Membrane Stress S_{mx} - Saddle

Figures 24 and 25 represent membrane stresses $S_{m\phi}$ at Edge 1, at Edge 2, and at the Saddle Center. The change in saddle angle does not seem to effect these stresses. The maximum stresses are compressive and occur at $\emptyset$ = 105 degrees for all cases.

The analysis presented by reference (3) does not concern itself with the membrane stresses in the vicinity of the saddle-shell interface. Hence these results are not compared. The Figures 22 through 25 represent the stresses obtained by superimposing all the membrane stresses induced by horizontal and pressure loads. The trend of the over-all shape of the curves is not clearly understood.

Displacements

Figures 26 and 27 show radial deformations for 140 degrees and 180 degrees saddle angles respectively. As shown in the figures, the effect of saddle height is small and less critical than the effect of the saddle angles.

The outward radial deformation below the horn of the saddle is positive, which implies that the shell bulges out of the saddle. In the analysis presented by reference (3), the shell overlaps the saddle because of the rigidity of the saddle. However, in this case, since the vessel and the shell are welded, they both deform together, which would reduce the maximum stresses at the saddle-shell interface. The bulging of the shell increases sharply with small increments of $\emptyset$ and reaches a maximum at 15 degrees above the saddle angle.

Figures 28 and 29 show tangential displacements for specified saddle heights and saddle angles. The tangential displacement is assumed to be positive in the direction of increasing $\emptyset$ as shown in

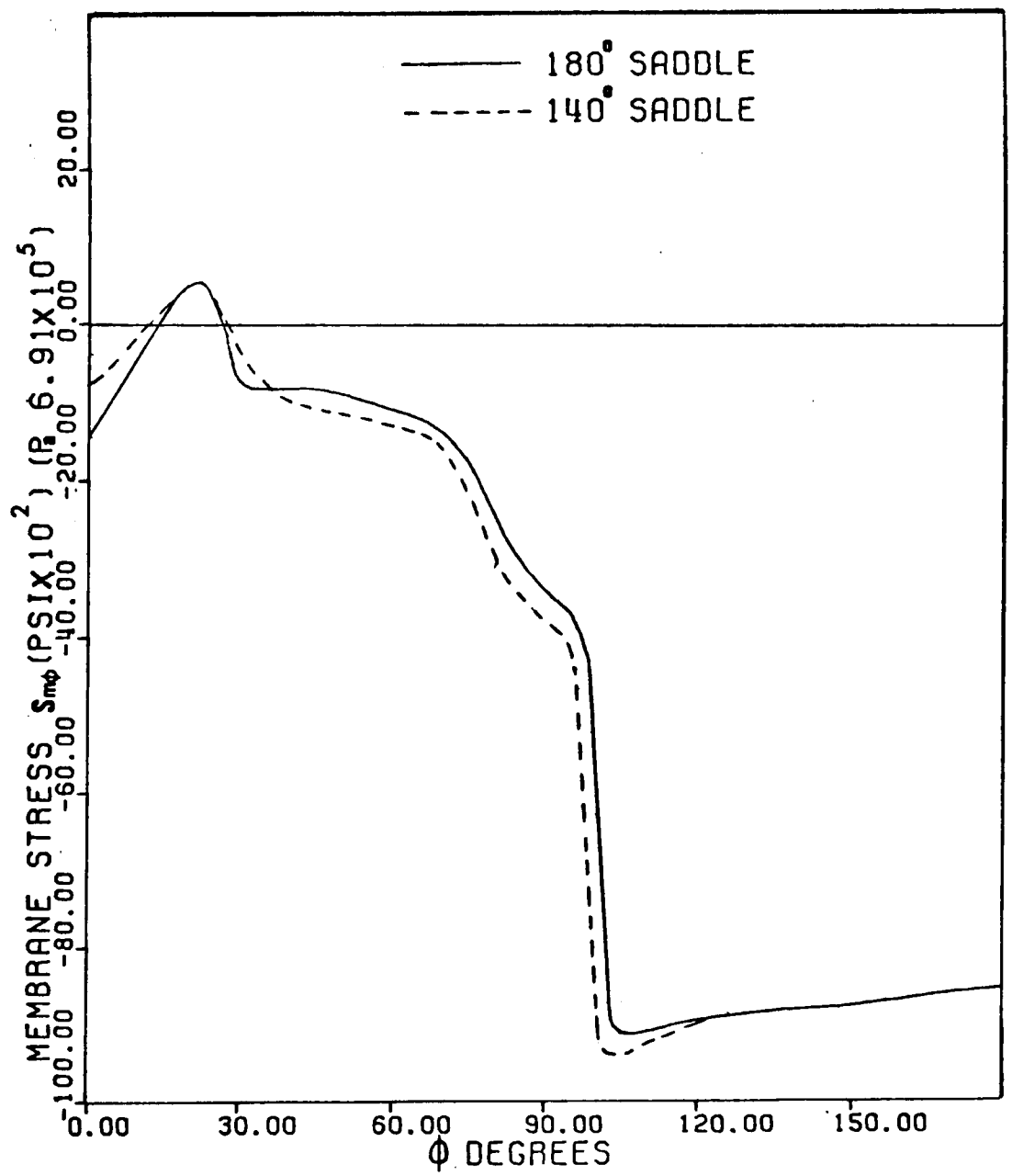

Figure 24. Membrane Stress $S_{m\phi}$ - Saddle Center

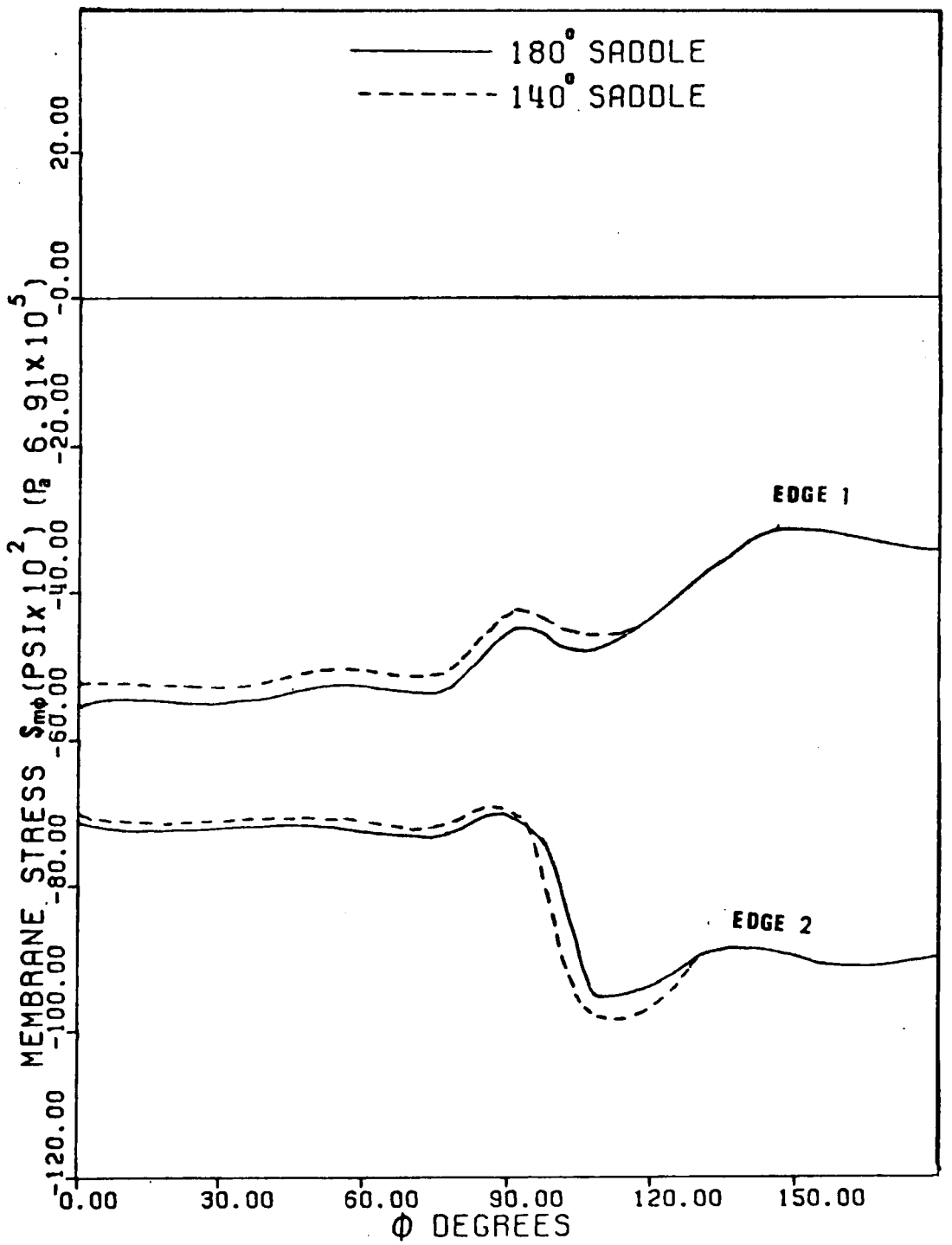

Figure 25. Membrane Stress $S_{m\phi}$

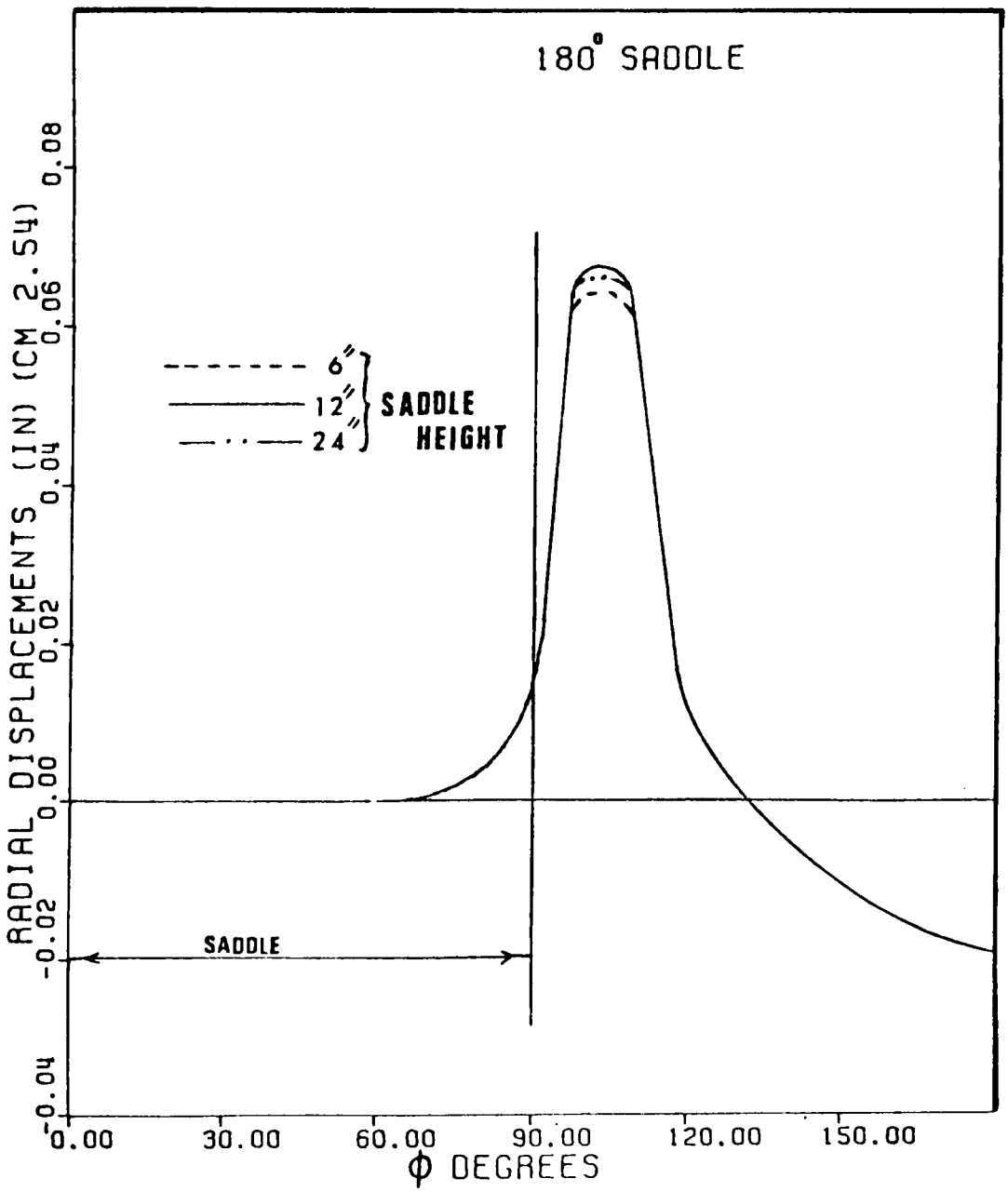

Figure 26. Radial Displacements - 180° Saddle

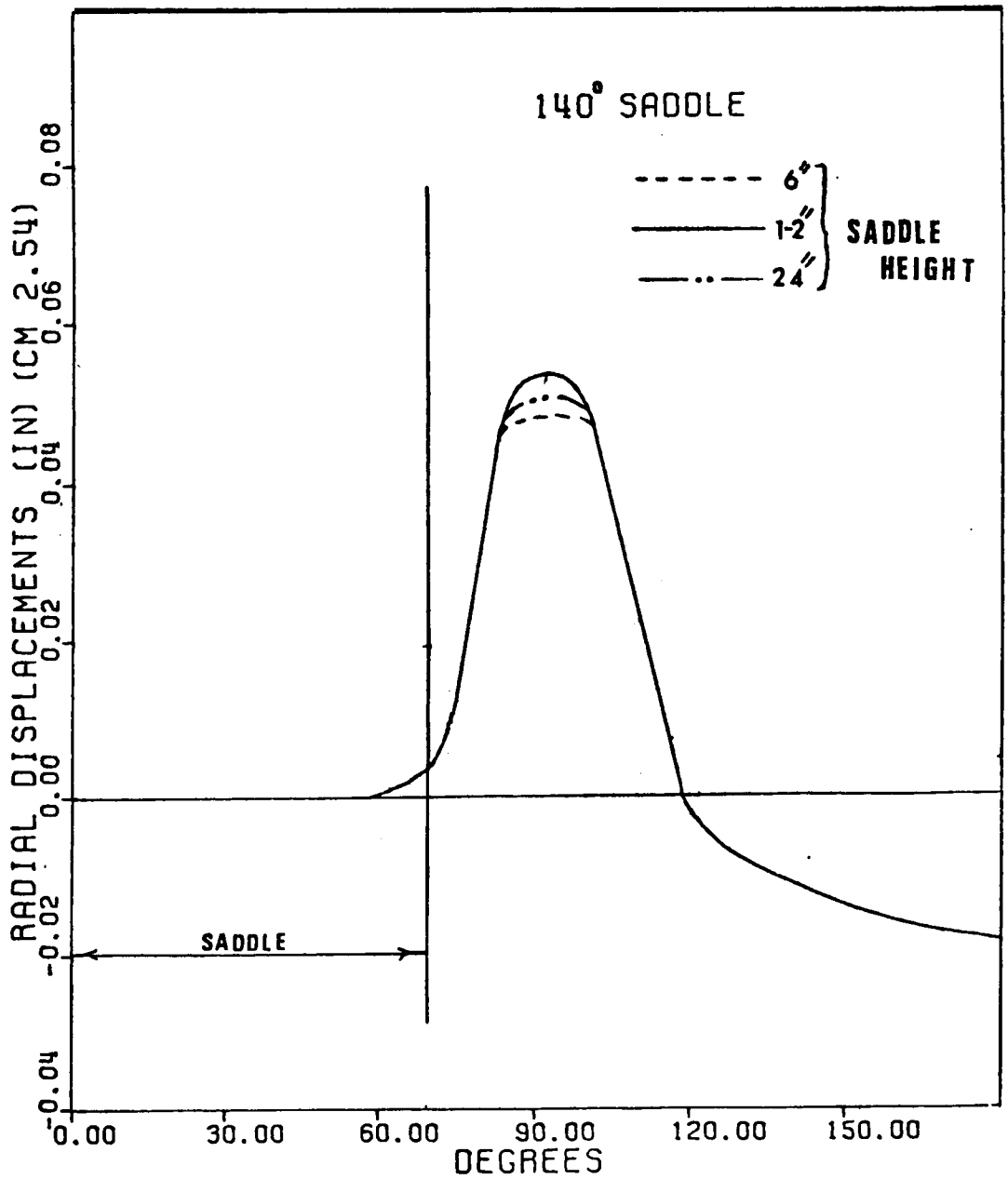

Figure 27. Radial Displacements - 140° Saddle

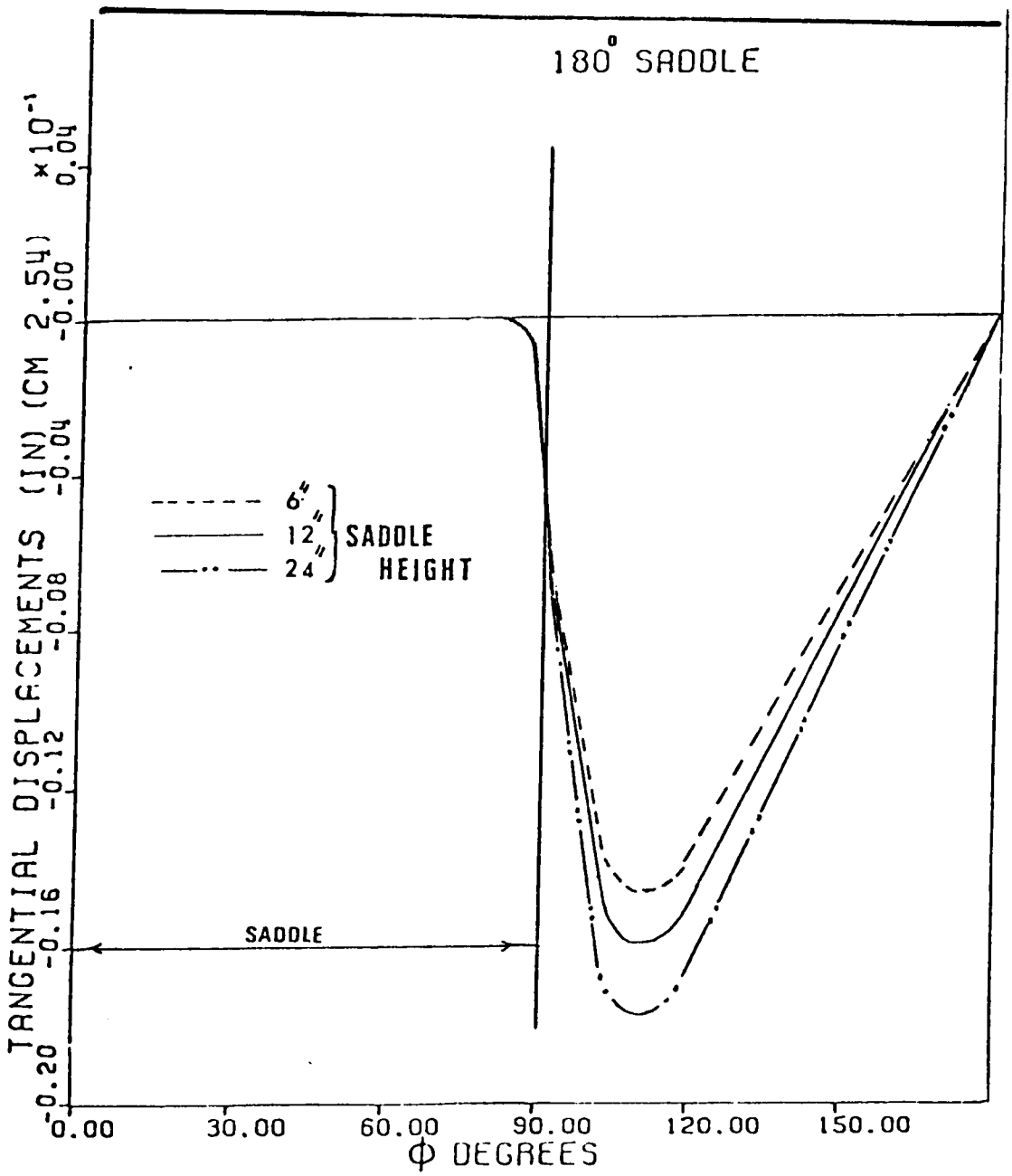

Figure 28.. Tangential Displacements - 180° Saddle

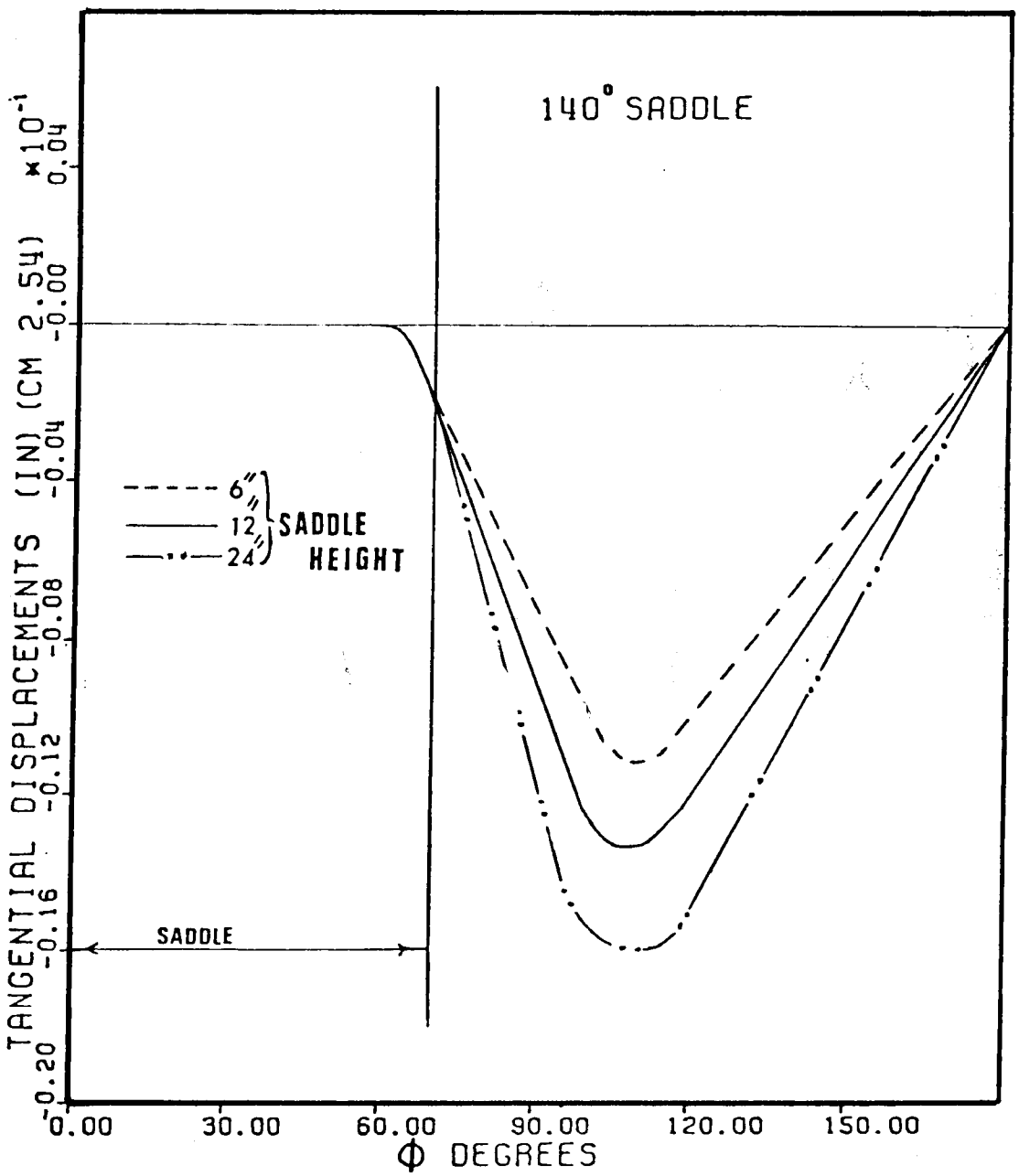

Figure 29. Tangential Displacements - 140° Saddle

Figure 17 (see page 38). The effect of saddle height is considerable in both cases. The two figures coincide reasonably when considering the over-all shape. However, the 180 degree saddle angle has a more pronounced effect than the 140 degree saddle angle. The maximum tangential displacement occurs at ϕ = 105 degrees for both the saddle angles and for the saddle heights.

Principal Stresses in the Ribs

To check the validity of the assumption that the outer rib can be treated as a beam, it was compared to the inner rib. It was found that the membrane stresses S_{mx} and S_{my} are linearly distributed about the neutral axis and the shear stress S_{xy} is maximum at the neutral axis as the common beam theory predicts. The ratio of span to depth for the outer rib is larger than the inner rib. Hence it can be concluded that the assumption for the outer rib is quite justified. The center rib is too short and does not follow the common beam theory; therefore it cannot be treated as a beam.

The bending stresses increase as the saddle height is increased. However, they are very small as compared to membrane stresses and can be neglected in the analysis. The membrane stress S_{mx} induced by the horizontal loads increases as the saddle height increases. But the membrane stress S_{my} and shear stress decrease. There is no apparent change in membrane stress as a function of saddle angle when subjected to horizontal loads.

The same problem was analyzed by ignoring the center rib. It was found that the loads were distributed between the inner and outer ribs. However, the stresses induced in the inner rib exceed the yield stress

of the material. This implies that it is always advantageous to put a center rib stiffener in the saddle.

The maximum stress occurs at both ends at the bottom of the rib where the anchor bolts are usually located. They are compressive toward the vessel head. Figure 30 shows that most of the load is taken up by the center rib. The stresses in the center rib are high and exceed the yield stress of the material. These high stresses could cause local buckling in the center rib since there is no lateral support to resist the load.

Anchor Bolt Forces

Figure 31 represents tension per anchor bolt at specified positions of the ribs for 180 degree and 140 degree saddle angles.

It is found that tension induced by horizontal loads has a more severe effect on the anchor bolts which are located at the center rib. An anchor bolt 1-1/2 inches in diameter is required to resist a tension of 26,000 lbs. The tension induced by horizontal loads indreases as the saddle height is increased. However, it does not seem to be a function of saddle angle.

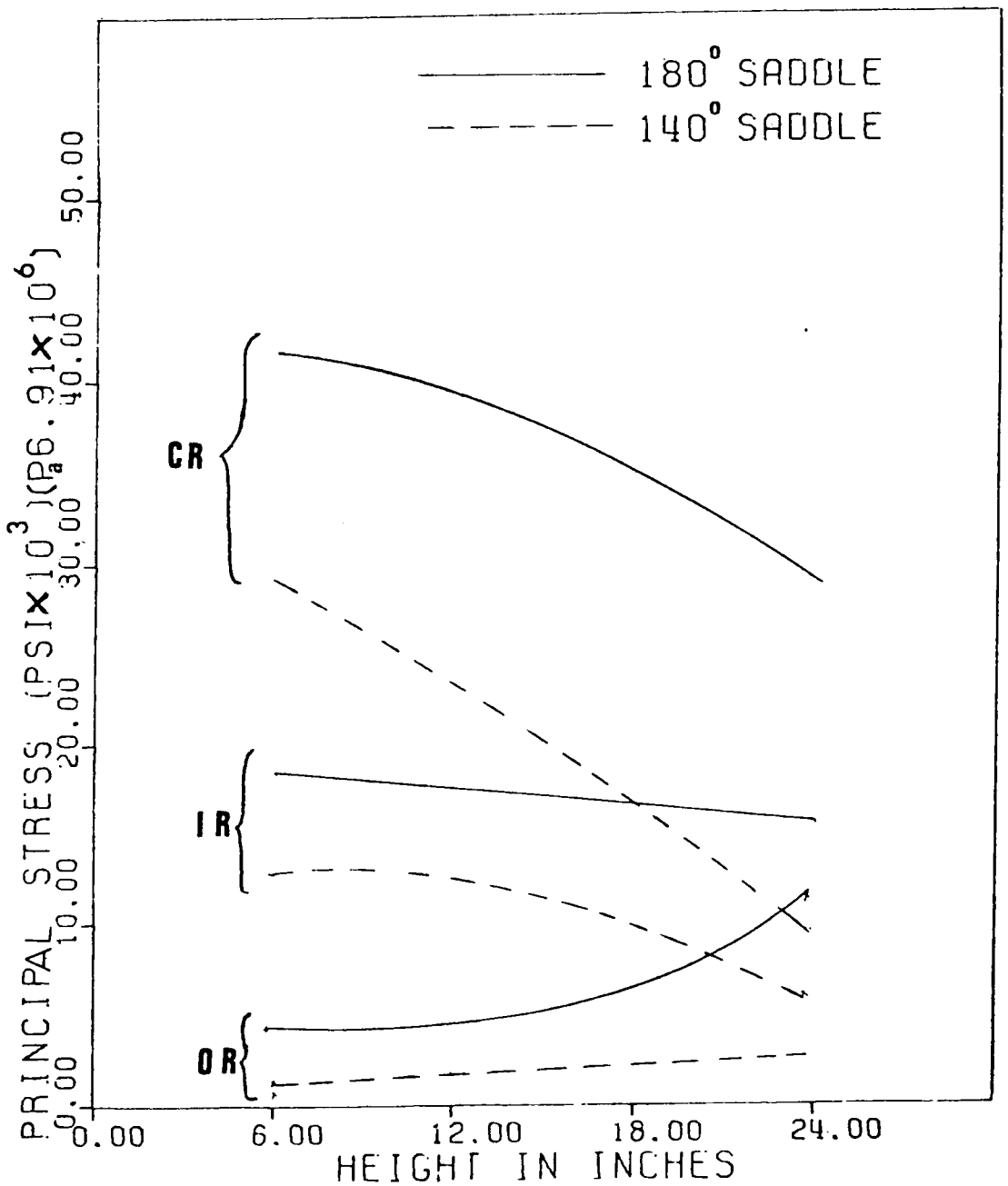

Figure 30. Principal Stresses in Ribs

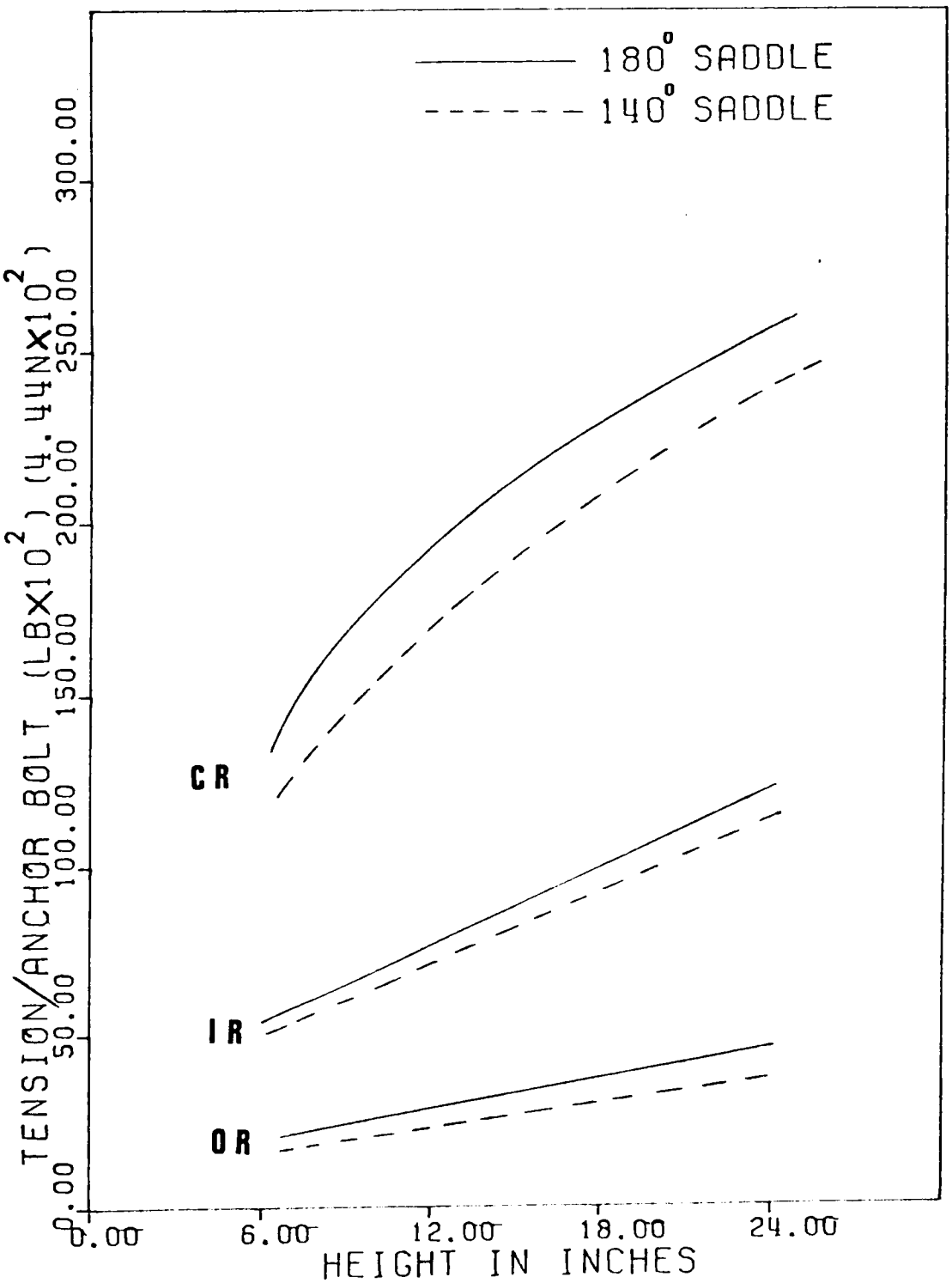

Figure 31. Tension in Anchor Bolts

CHAPTER V

CONCLUSION

The results of the stress analysis of a two saddle supported, horizontal pressure vessel permits the following conclusions.

In references (4) and (5), the interface pressure is assumed to vary in the circumferential direction, but is held constant in the longitudinal direction. The present author shows that the assumption of constant saddle-shell interface pressure in the longitudinal direction is too restrictive. When this constraint is removed, the stress distribution in the vessel near the saddle is significantly altered as compared to the results in references (4) and (5). In addition, reference (5) consistently gives stresses much higher than those predicted by the present investigation and references (3) and (4). Since the finite element results more nearly agree with both references (3) and (4), the present author is inclined to feel that his results are more nearly correct than the results of reference (5). In their paper, Lakis and Dore (4) indicate that they are currently developing the general theory necessary to calculate the correct interface pressure distribution. When these results become available, it will be interesting to compare their results with the present results obtained from a finite element solution.

The present results also indicate that the maximum stresses in the vicinity of the saddle-shell juncture are approximately the same for both the welded and unwelded cases. Furthermore, Zick's analysis gives maximum circumferential bending stresses that closely agree with the

finite element analysis for the particular vessel analyzed. This implies that designers could use Zick's analysis for both the welded and unwelded cases. However, the reader is cautioned that additional parametric studies should be undertaken before any broad conclusions can be made as to the applicability of Zick's analysis to the welded case for other pressure vessels.

The results pertaining to the study of the stress analysis of the support design for a horizontal pressure vessel subjected to seismic as well as pressure loads permit the following conclusions.

The analysis presented by reference (3) to calculate the maximum circumferential bending stresses at the saddle-shell interface is well within the agreement of the finite element analysis for the vessels analyzed. It can be concluded that even though Zick's analysis does not address itself to seismic loads it is adequate to design the horizontal pressure vessels, when subjected to horizontal and pressure loads. In this case it is recommended that the reader perform additional parametric studies in order to verify the adequacy of Zick's analysis for other pressure vessels.

The forces in the anchor bolts are not equally distributed. They are subjected to high tensile loads in addition to moderate shear loads. If a design includes improperly sized anchor bolts, there is a possibility that they will fail under these loads and allow an undesirable redistribution of stress in the support structure which could, in turn, cause a support failure.

The principle stresses in the center rib are compressive toward the vessel head and exceed the yield stress of the material. The high

stresses imply that these ribs may be subjected to local lateral buckling unless appropriately stiffened. However, the study of buckling instability was not a part of this investigation.

These results indicate that considerable attention must be directed toward the development and analysis of saddle designs capable of resisting seismic loading. The author feels that a logical next stage of research effort in this area is derivation of a codified procedure for engineers to follow in designing "standard" saddle supports for horizontal pressure vessels which must be capable of resisting horizontal seismic loading.

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APPENDIX

ZICK'S ANALYSIS

The analysis considers horizontal pressure vessels on two saddle supports as a beam with the following assumptions:

i) half the load is taken by each support since they are placed symmetrically.

ii) the vessel is assumed to be completely filled with water.

iii) for simplicity, the depth of the head is assumed to be zero, i.e., H = 0.

iv) all the loads, acting on the vessel, are combined vectorially to find the saddle reaction.

v) the external pressure, wind load and other impact loads are neglected in this analysis.

The analysis is capable of predicting the following stresses at specified locations.

i) Longitudinal Bending Stress:
   a) at the Saddles
   b) at the Midspan

ii) Tangential Shear Stress:
   a) in the Shell

iii) Circumferential Bending Stress:
   a) at the horn of the Saddle
   b) at the bottom of the Shell

Input Data

For convenience of the reader, the following input data are reproduced from Chapter IV, Section II.

L = 240" tangent to tangent length

A = 12" distance from tangent line of the head to the center of the saddle

b = 8" width of the saddle

P = 24" psi maximum internal design pressure

R = 57" outside radius of the shell

t_w = .125" thickness of the wear plate

t_s = 0.25" thickness of the shell

ϕ = 140 degrees saddle angle

S_a = 12,600 psi allowable stress of the material

D_s = 0.283 lbs/in^3 density of steel

D_w = 0.03611 lbs/in^3 density of water

Q = resultant load on one saddle in lbs.

Calculations

Weight of the water:

W_w = (volume (density)

= (3.14 R^2 L) (D_w)

= (3.14) (57) (57) (24) (.03611)

= 88458 lbs

Weight of the shell:

W_s = (volume) (density)

= (3.14) (57) (2) (240) (.25) (.283)

= 6081 lbs

Total weight:

$$W_t = W_w + W_s$$

$$= 88458 + 6081$$

$$= 94539 \text{ lbs}$$

Since only one quarter of the vessel is analyzed the reaction in each saddle will be

$$Q = W_t/4 = 23635 \text{ lbs.}$$

Longitudinal Bending Stress (S_1)

Stress at the saddles:

$$S_1 = Q\,A(1-(1-A/L + R^2/2\,A\,L))/K_1\,R^2\,t_s$$

Substituting the value of $K_1 = 0.443$ from the reference (6) in to the above equation

$$S_1 = 405 \text{ psi}$$

Stress at midspan:

$$S_1 = \frac{\frac{Q\,L}{4}\left(1 + \frac{2\,R^2}{L^2} - \frac{4\,R}{L}\right)}{3.14\,R^2\,t_s}$$

$$= 507 \text{ psi}$$

When in tension, S_1 should be less than the allowable stress times the joint efficiency of the girth seam minus the stress due to internal pressure.

The allowable tension stress = 12,600 x 0.85 = 10710 psi. The stress due to internal pressure = $\frac{P\,R}{2\,t_s}$ = 2736 psi. Therefore, net allowable stress is 10710 - 2736 = 7974 psi 1463 $\leq$ 7974

In compression the maximum allowable stress is given by the equation

$$S_1 \leq (E/29)(t/R(2-(2/3)(100)(t/R)$$

$$405 \leq 7748$$

Hence it is acceptable.

Tangential Shear Stress (S_2)

$$S_2 = \frac{K\ Q}{R\ t_s}$$

$$= 982 \text{ psi}$$

S_2 should be less than 0.8 times the maximum allowable stress of the material

$$982 \leq 0.8 \times 12{,}600 = 10080$$

Circumferential Stress (S_4)

Stresses at the horn of the saddle

$$S_4 = -\frac{Q}{4\ t_s(b + 1.56(R\ t_s)^{1/2}} - \frac{12\ K_6\ Q\ R}{L\ t_s^2}$$

$$= -13557 \text{ psi}$$

Since the circumferential stress exceeds the maximum allowable stress a wear plate is required

$$t_s = t_s + t_w$$

$$= 0.375"$$

Recalculating the circumferential stress with the above thickness

$$S_4 = -6970 \text{ psi}$$

Stress at the bottom of the shell (S_5)

$$S_5 = - \frac{K_7 \, Q}{t_s (b + 1.56 \, (R \, t_s)^{1/2})}$$

$$= - 2887 \text{ psi}$$

The circumferential stresses are less than one half the compression yield point.

VITA

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