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## A COMPARISON OF THREE POINT AND FOUR POINT LOADING IN ELASTIC-PLASTIC BENDING OF BEAMS

R. Vincent Milligan

June 1981



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20. ABSTRACT (CONT'D)

removal of permanent deflections with the possible elimination of hot straightening in some instances. The concept of distribution of plastic deformation is put forth as a possible explanation of the advantage of one method over the other.

## TABLE OF CONTENTS

	<u>Page</u>
INTRODUCTION	1
RESULTS AND DISCUSSION	6
CONCLUSIONS	9
REFERENCES	11
APPENDIX A	A-1
APPENDIX B	B-1

## ILLUSTRATIONS

1. Load and Moment Diagrams for Three Point Bending.	2
2. Load and Moment Diagrams for Four Point Bending.	4
3. Load vs. Deflection.	12
4. Moment of Inertia vs. Deflection.	13
5. Moment of Inertia vs. Deflection.	14
6. Ratio of Four Point Deflection to Three Point Deflection.	15
7. Ratio of Four Point Deflection to Three Point Deflection vs. Outside Radius.	16
8. Maximum Deflection Under Load vs. Length of Plastic Zone.	17
A1. Strain vs. Depth of Elastic-Plastic Interface.	A-1
B2. Strain vs. Depth of Cross-Section.	B-1

## INTRODUCTION

One of the questions that arises in the area of straightening components deals with the magnitude of deflections and the corresponding strains which could adversely affect material properties and limit subsequent usage of the component. An equally pertinent question centers on the need for hot straightening to eliminate excessive permanent deflections or bows in components due to processing operations. With these questions forming a backdrop, the writer performed an analysis comparing three and four point bending from the standpoint of seeing whether larger permanent deflections could be removed using the four point method and thus possibly obviating the need for hot straightening. It was hoped that this could be done without causing additional material degradation as a result of straining the material into the plastic region.

The intent of this report is to present the results of a theoretical elastic-plastic analysis as a means of comparing the two methods.

The theory used as a basis for the calculations in this report is essentially the same as developed by Seely and Smith.<sup>1</sup> We first consider the case for a concentrated load at midspan as shown in Figure 1(a). Using the dummy load method, which is practically the same as the principle of virtual work, the deflection can be calculated from the integral

$$\delta = \int m d\alpha \quad (1)$$

---

<sup>1</sup>F. B. Seely and J. O. Smith, Advanced Mechanics of Materials, 2nd Ed., Wiley and Sons, 1952.

where  $m$  is the moment due to the dummy load and  $d\alpha$  the relative rotation of one plane of a cross section with respect to another a distance  $dx$  apart. For the linear region  $d\alpha$  is equal to  $(M/EI) dx$ , where  $M$  is the moment due to the real load,  $E$  is the modulus of elasticity, and  $I$  is the moment of Inertia. We first integrate through the elastic region from 0 to  $\lambda$ , where  $\lambda$  is the distance from the left support to the end of the elastic region. We then have

$$\delta_1 = \int_0^\lambda \frac{Mm}{EI} dx \quad (2)$$

The limit  $\lambda$  depends on the loading conditions and can be obtained by ratio from the moment diagram. Referring to Figure 1(b)

$$\frac{\lambda}{M_y} = \frac{l/2}{M_{max}} \quad (3)$$

hence

$$\lambda = \frac{M_y}{M_{max}} (l/2) \quad (4)$$

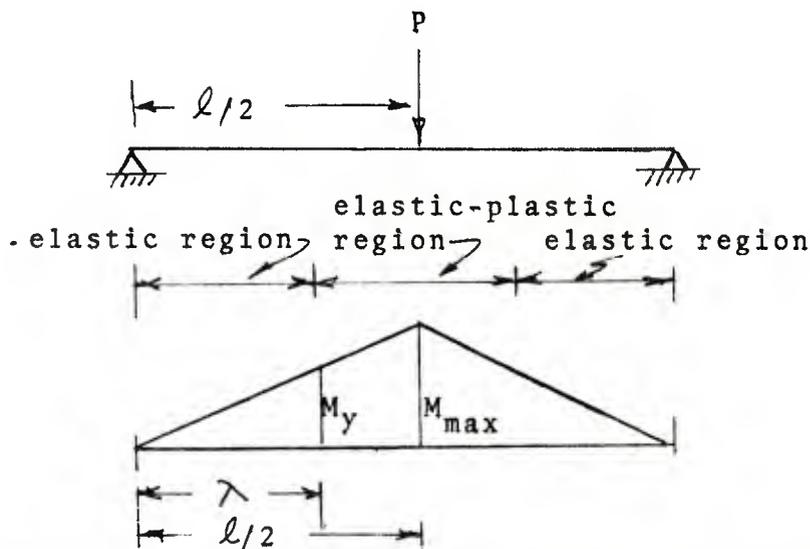


Figure 1. Load and Moment Diagrams for Three Point Bending.

As the moment increases from the yield moment  $M_y$  to the fully plastic moment  $M_{fp}$ ,  $\lambda$  decreases, indicating that the length of the elastic-plastic zone increases.

We next integrate through the elastic-plastic zone from  $\lambda$  to  $\ell/2$ . For the elastic-plastic region  $d\alpha$  now takes the form of

$$d\alpha = \frac{M_y}{EI} \frac{n^{1/2} dx}{\sqrt{K - \frac{M_p}{M_y}}} \quad (5)$$

One can refer to the Appendix for the development of this expression. Hence,

$$\delta_2 = \int_{\lambda}^{\ell/2} \frac{M_y}{EI} \frac{n^{1/2} m dx}{\sqrt{K - \frac{M_p}{M_y}}} \quad (6)$$

The total deflection can then be obtained by adding deflections  $\delta_1$  and  $\delta_2$  and multiplying by two since we have symmetrical loading.

For a simply supported beam having a concentrated load,  $M = Px/2$  and  $m = x/2$  for the region  $0 < x < \ell/2$ . Hence:

$$\delta_1 = \int_0^{\lambda} \frac{P}{4} \frac{x^2}{EI} dx \quad (7)$$

$$\text{and } \delta_2 = \frac{M_y}{EI} n^{1/2} \int_{\lambda}^{\ell/2} \frac{x}{2} \frac{1}{\sqrt{K - \frac{Px}{2M_y}}} dx \quad (8)$$

Inserting the factor two previously mentioned, we obtain:

$$\delta = \delta_1 + \delta_2 = \frac{P}{2EI} \int_0^{\lambda} x^2 dx + \frac{M_y}{EI} n^{1/2} \int_{\lambda}^{\ell/2} \frac{x}{\sqrt{K - \frac{Px}{2M_y}}} dx \quad (9)$$

Integrating and inserting limits, we get the equivalent expression given by Seely and Smith<sup>1</sup>

$$\delta = \frac{P\ell^3}{48EI} \frac{M_y^3}{M_{\max}^3} \left[ 1 + 2n(2K+1) - 2n^{1/2} \left( 2K + \frac{P\ell}{4M_y} \right) \left( K - \frac{P\ell}{4M_y} \right)^{1/2} \right] \quad (10)$$

It is readily apparent that this expression degenerates to the elastic equation when  $M_{\max} = M_y$ . That is,

$$\delta = \frac{P\ell^3}{48EI} \quad (11)$$

A computer program was written for equation (10) and given the acronym DEFL. The program has a Do Loop on P. The results shown graphically were obtained from this program.

We next consider the case for two concentrated loads symmetrically located relative to the midspan as shown in Figure 2(a).

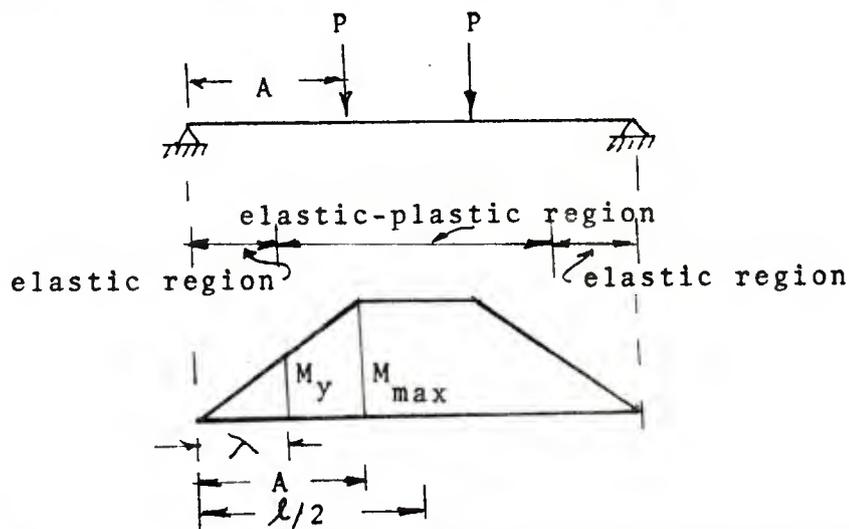


Figure 2. Load and Moment Diagrams for Four Point Bending.

<sup>1</sup>F. B. Seely and J. O. Smith, Advanced Mechanics of Materials, 2nd Ed., Wiley and Sons, 1952.

In this case, we have to integrate through three regions, 0 to  $\lambda$ ,  $\lambda$  to A, and A to  $\ell/2$ .

$$\delta_1 = \int_0^\lambda \frac{Mdx}{EI} = \frac{1}{EI} \int_0^\lambda Px\left(\frac{x}{2}\right)dx \quad (12)$$

where the limit  $\lambda$  by ratio is equal to  $\frac{M_y}{M_{\max}} A$  as can be noted from Figure 2(b).

$$\delta_2 = \int_\lambda^A \frac{M_y}{EI} \frac{n^{1/2} dx}{\sqrt{K - \frac{M_y}{M_y}}} = \frac{M_y}{EI} n^{1/2} \int_\lambda^A \frac{x/2}{\sqrt{K - \frac{Px}{M_y}}} dx \quad (13)$$

and

$$\delta_3 = \int_A^{\ell/2} \frac{M_y}{EI} \frac{n^{1/2} dx}{\sqrt{K - \frac{M_{\max}}{M_y}}} = \frac{M_y n^{1/2}}{EI} \frac{1}{\sqrt{K - \frac{M_{\max}}{M_y}}} \int_A^{\ell/2} \left(\frac{x}{2}\right) dx \quad (14)$$

Evaluating these integrals, and inserting the factor of two because of symmetry and integrating over half the length, the maximum deflection at  $\ell/2$  becomes

$$\delta = \delta_1 + \delta_2 + \delta_3 \quad (15)$$

$$\delta = \frac{PA^3}{3EI} \frac{M_y^3}{M_{\max}^3} + \frac{2}{3} \frac{M_y^3}{EI} \frac{n^{1/2}}{P^2} \left[ -\left(2K + \frac{PA}{M_y}\right) \left(K - \frac{PA}{M_y}\right)^{1/2} + (2K+1)(K-1)^{1/2} \right] + \frac{M_y}{EI} \frac{n^{1/2}}{\sqrt{K - \frac{PA}{M_y}}} \left[ \frac{\ell^2}{8} - \frac{A^2}{2} \right] \quad (16)$$

A computer program was written for equation (16) and given the acronym SSDEFL2P. The program has a Do Loop on P. The results shown graphically were obtained from this program.

If we specialize this equation for  $A = \ell/2$ , the second and third terms drop out and we get

$$\delta = \frac{P\ell^3}{24EI} \quad (17)$$

This differs from equation (11) by a factor of two since we have two loads acting at  $\ell/2$ .

#### RESULTS AND DISCUSSIONS

As a means of comparing three point bending with four point bending, the following cases were calculated using the previously mentioned computer programs with the following input data:

$$R_0 = 3.0, 3.5, 4.0, 4.5 \text{ inches}$$

$$R_1 = 2.0 \text{ inches}$$

$$E = 30 \times 10^6 \text{ psi}$$

$$\sigma_y = 160 \text{ Ksi}$$

$$\ell = 60 \text{ inches}$$

$$A = 15, 20, 25 \text{ inches}$$

The load  $P$  was located at  $\ell/2$  for the case of three point bending. This gave a total of 16 load-deflection curves. An example for the case of  $R_0 = 3.0$  inches with  $A = 15, 20,$  and  $25$  inches is shown in Figure 3. Assuming linear elastic unloading, the permanent deflections were obtained by projecting downward from the maximum deflection at maximum load, to the deflection axis with a line parallel to the initial elastic loading line. It should be pointed out that load  $P$  which would be applied to the loading fixture to accomplish the four point loading is actually equal to  $2P$  and is consequently twice the value of  $P$  applied in the three point case. The same is also true for all

subsequent diagrams and figures comparing three point and four point loading. From these 16 curves we constructed two others, Figures 4 and 5. Figure 4 is a plot of moment of inertia (I) vs. deflection for the three point loading and three different values of A for the four point loading. The values of I for four different outside radii are marked. The solid lines show the maximum deflection under load and the dashed lines are for the permanent deflections. The plasticity condition is such that the elastic-plastic interface has moved a distance equal to 75 percent of the outside radius from the outside fiber toward the neutral axis. Equation 35 of reference 2 gives a relationship between the outside radius  $R_0$ , the distance from the neutral axis to the elastic-plastic interface  $\rho$ , and the yield strain,  $\epsilon_y$ . That is,

$$\epsilon_{\max} = \frac{R_0}{R_0 - \rho} \epsilon_y$$

For the 75 percent plasticity condition used for comparison purposes for the two methods,  $R_0$ ,  $\rho$ , and  $\epsilon_y$  would be the same, therefore the maximum strains would be the same for both bending methods. The same reasoning would hold for the nearly 100 percent plasticity condition also considered.

Figure 5 is the same plot as Figure 4 except the plasticity condition is such that the elastic-plastic interface has moved a distance equal to nearly 100 percent of the outside radius. Table I gives a summary of the various geometrical parameters, loads, moments, and deflections.

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<sup>2</sup>R. V. Milligan, "Moment-Strain Relationships in Elastic-Plastic Bending of Beams," to be published.

Figure 6 was obtained by taking ratios of maximum deflections under load for the four point case to the three point case for a constant R. For the 100 percent plasticity condition, the maximum ratio is 7.85 for  $A = 15$  and  $R_0 = 3.0$  while the minimum ratio is 3.25 for  $A = 25$  and  $R_0 = 4.5$ . For the 75 percent plasticity condition, the maximum ratio is 2.90 for  $A = 15$  and  $R_0 = 3.0$  while the minimum ratio is 1.61 for  $A = 25$  and  $R_0 = 4.5$ . It is thus evident for both plasticity conditions that the maximum ratios occur for the beam having a smaller cross section and as the loads get nearer to the supports.

Figure 7 is the same as Figure 6, except the deflection ratios are permanent deflections. Again the maximum ratio for the 100 percent plasticity condition occurs for  $A = 15$  and  $R_0 = 3.0$  while the minimum ratio occurs for  $A = 25$  and  $R_0 = 4.0$ . For the 75 percent plasticity condition, the maximum ratio occurs for  $A = 15$  at a value of  $R_0$  between 3.5 and 4.0.

In summary, the ratio of maximum deflections for the 100 percent plasticity condition run from a low of about 3.25 to a high of about 7.75. For the 75 percent plasticity condition they run from a low of about 1.75 to a high of about 3.75. The ratios are even greater for the permanent deflections. Here they run from a low of 6.25 to a high of 19 for the 100 percent condition and from about 2 to 8.75 for the 75 percent condition. It is thus very evident, based on the theory presented here, that four point bending can be a means of significantly increasing the deflections compared to three point bending, while at the same time keeping the strains and material degradation at the same level.

In an effort to propose a reason for the apparent advantage of four point bending, the author submits the concept of distributed plastic flow. Figure 8 is a plot of maximum deflection under load vs. length of plastic zone. The length of plastic zone can be determined by simple ratio as follows. For the four point case:

$$\frac{x}{M_y} = \frac{A}{M_{\max}} \quad \text{hence} \quad x = \frac{M_y}{P_{\max}A} \cdot A = \frac{M_y}{P_{\max}}$$

where  $s$  is the distance from the left support to the beginning of the plastic zone. For the three point case:

$$\frac{x}{M_y} = \frac{l/2}{M_{\max}} \quad \text{hence} \quad x = \frac{M_y}{M_{\max}} (l/2) = \frac{M_y}{P_{\max}/2}$$

Figure 8 shows sketches depicting the size of the elastic-plastic zones both for the three point case and three different values of  $A$  for the four point case. Obviously, as the loads move outward toward the supports, the elastic-plastic region becomes larger. The plot of  $\delta_{\max}$  vs. length of plastic zone is nearly linear. This tends to support the concept that as one distributes the plastic flow one can increase the deflection without appreciably increasing the maximum fiber strain with its probable consequence of degradation.

#### CONCLUSIONS

From the analysis presented it appears that larger deflections under load can be obtained and consequently larger permanent deformations can be removed by using the four point bending method compared with three point bending while keeping the maximum fiber strains the same in both cases.

The concept of distributed plastic flow appears to be a valid reason for the advantage of the four point method relative to the three point method.

## REFERENCES

1. F. B. Seely and J. O. Smith, Advanced Mechanics of Materials, 2nd Ed., Wiley and Sons, 1952.
2. R. V. Milligan, "Moment-Strain Relationships in Elastic-Plastic Bending of Beams," to be published.
3. J. H. Faupel, Engineering Design, Wiley and Sons, 1964, p. 395.

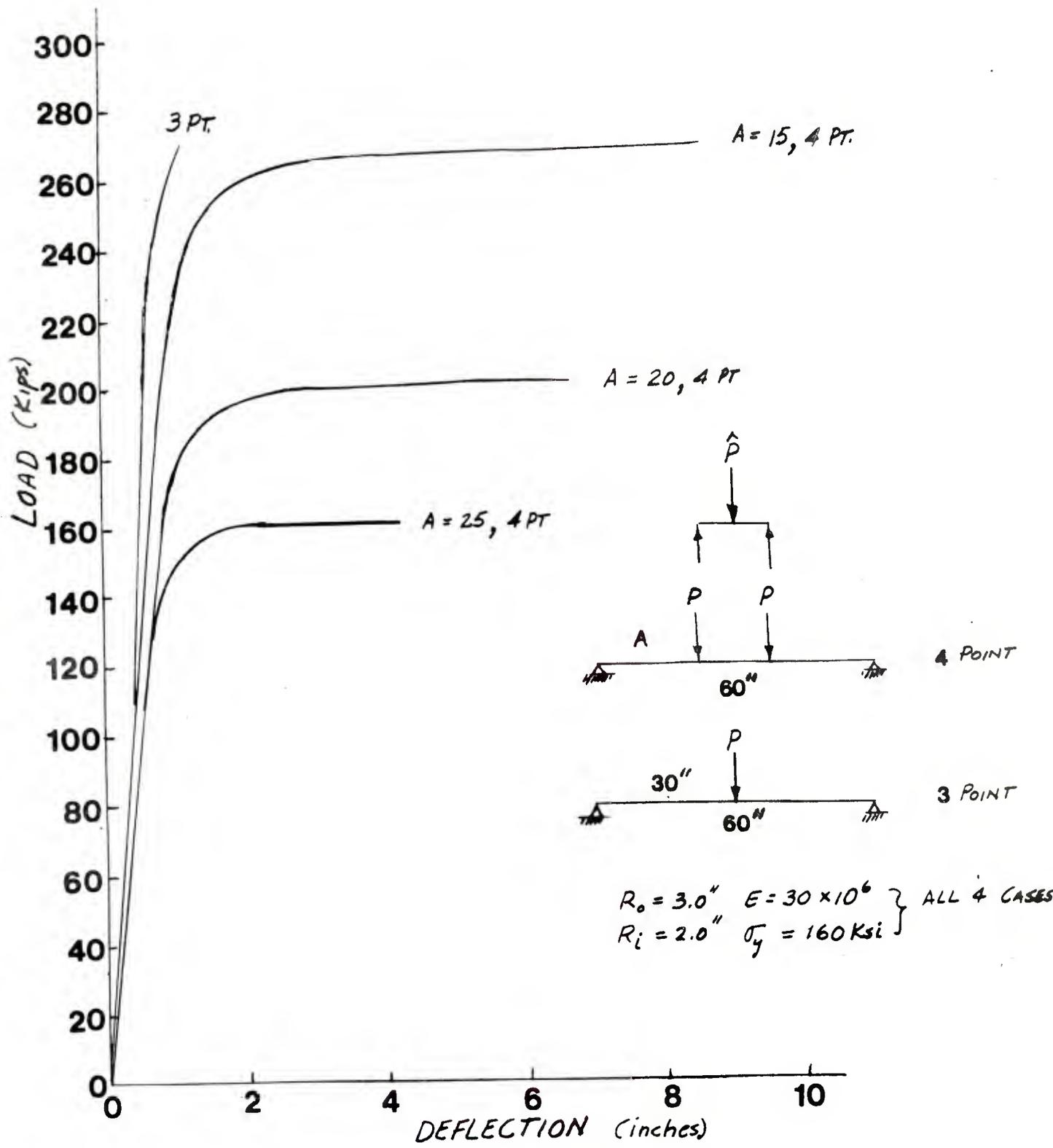
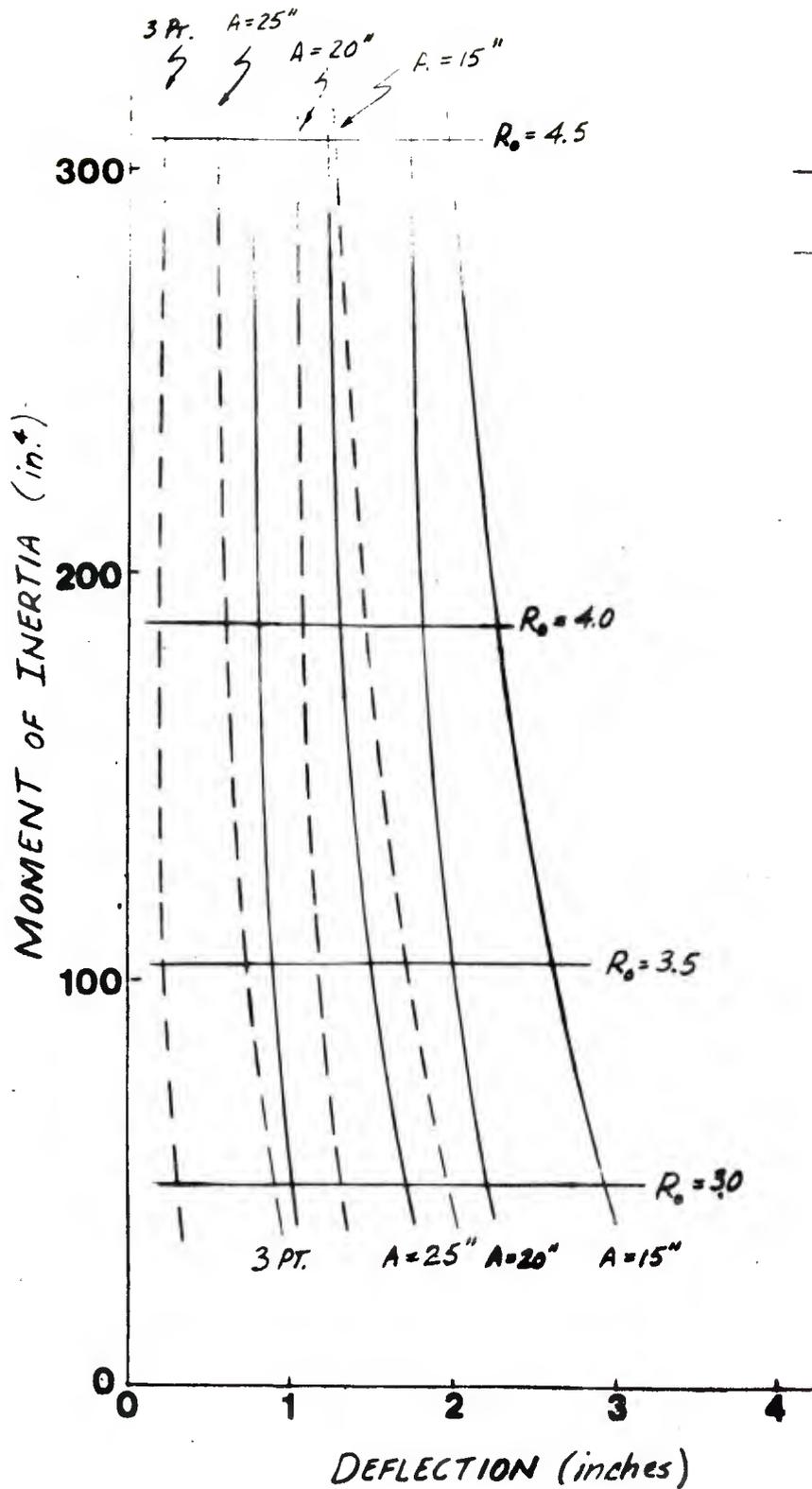
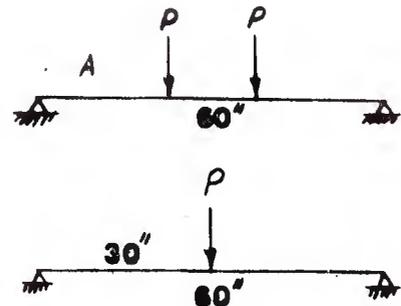


Figure 3. Load vs. Deflection.



—  $\delta_{MAX.}$   
 - - -  $\delta_{PERM.}$



$R_o = 3.0"$   $E = 30 \times 10^6$   
 $R_i = 2.0"$   $\sigma_y = 160 \text{ ksi}$

ALL 4 CASES

PLASTICITY CONDITION:

$$P = 0.75 R_o$$

Figure 4. Moment of Inertia vs. Deflection.

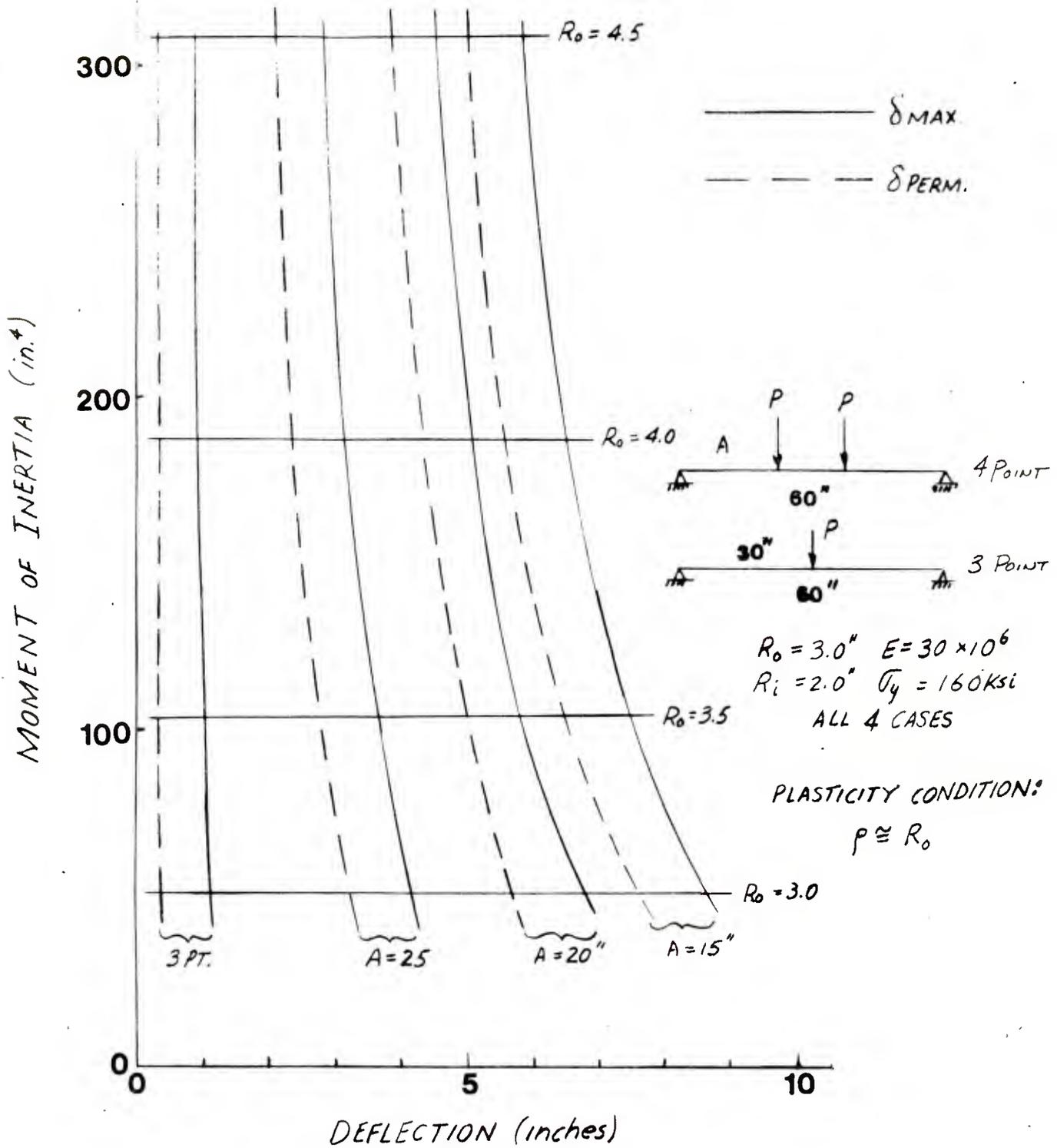


Figure 5. Moment of Inertia vs. Deflection

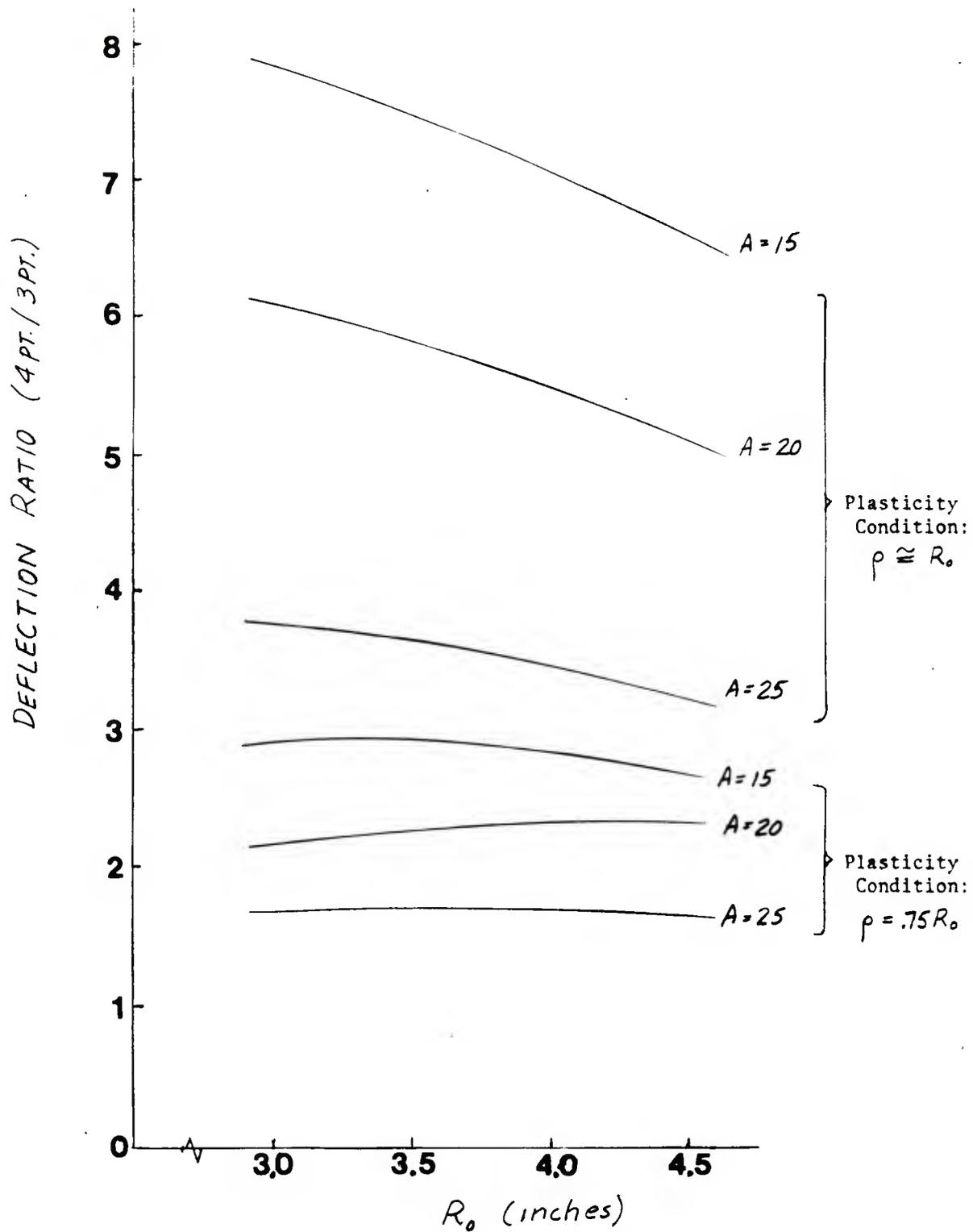


Figure 6. Ratio of Four Point Deflection to Three Point Deflection.

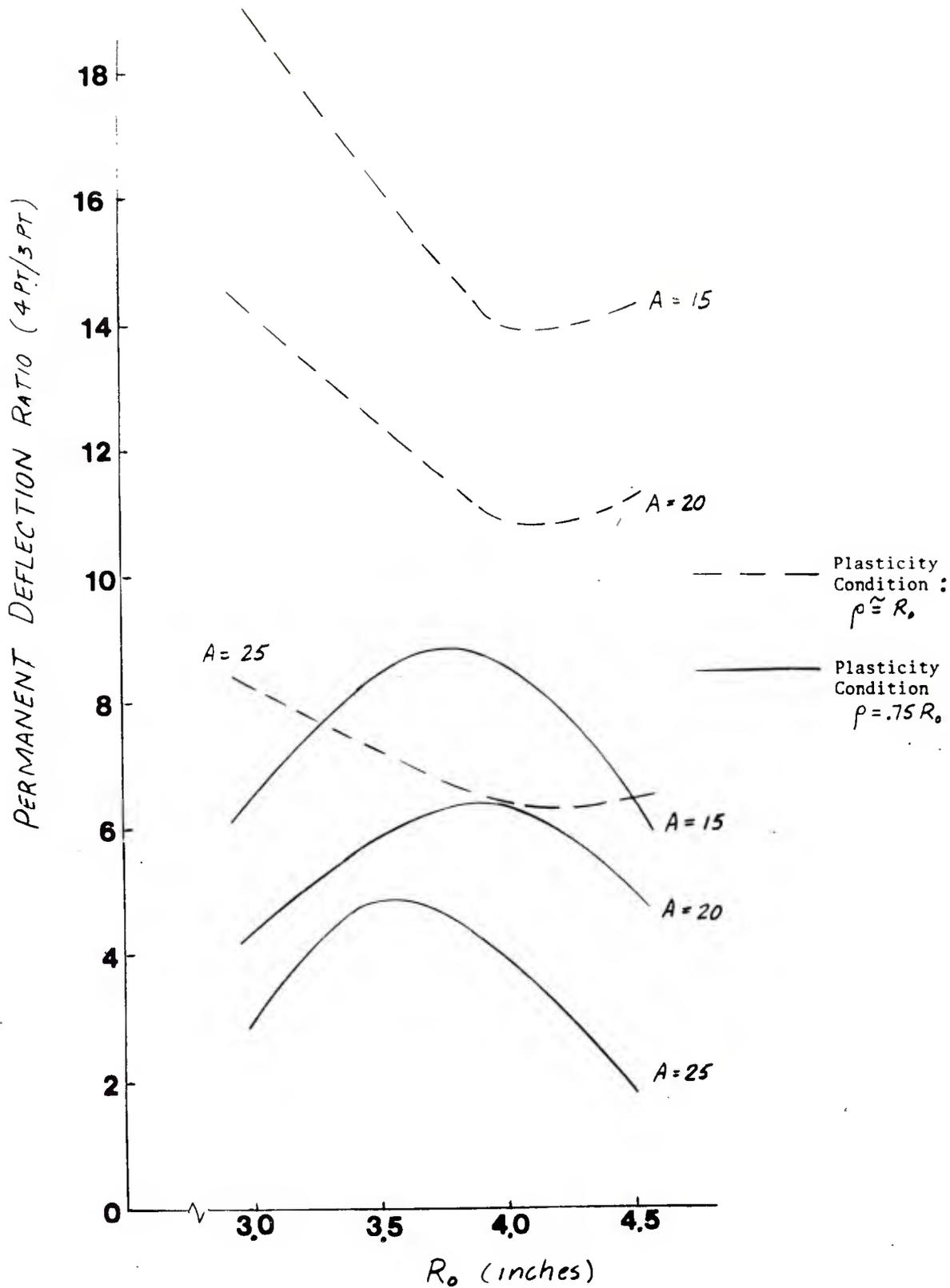


Figure 7. Ratio of Four Point Deflection to Three Point Deflection vs. Outside Radius.

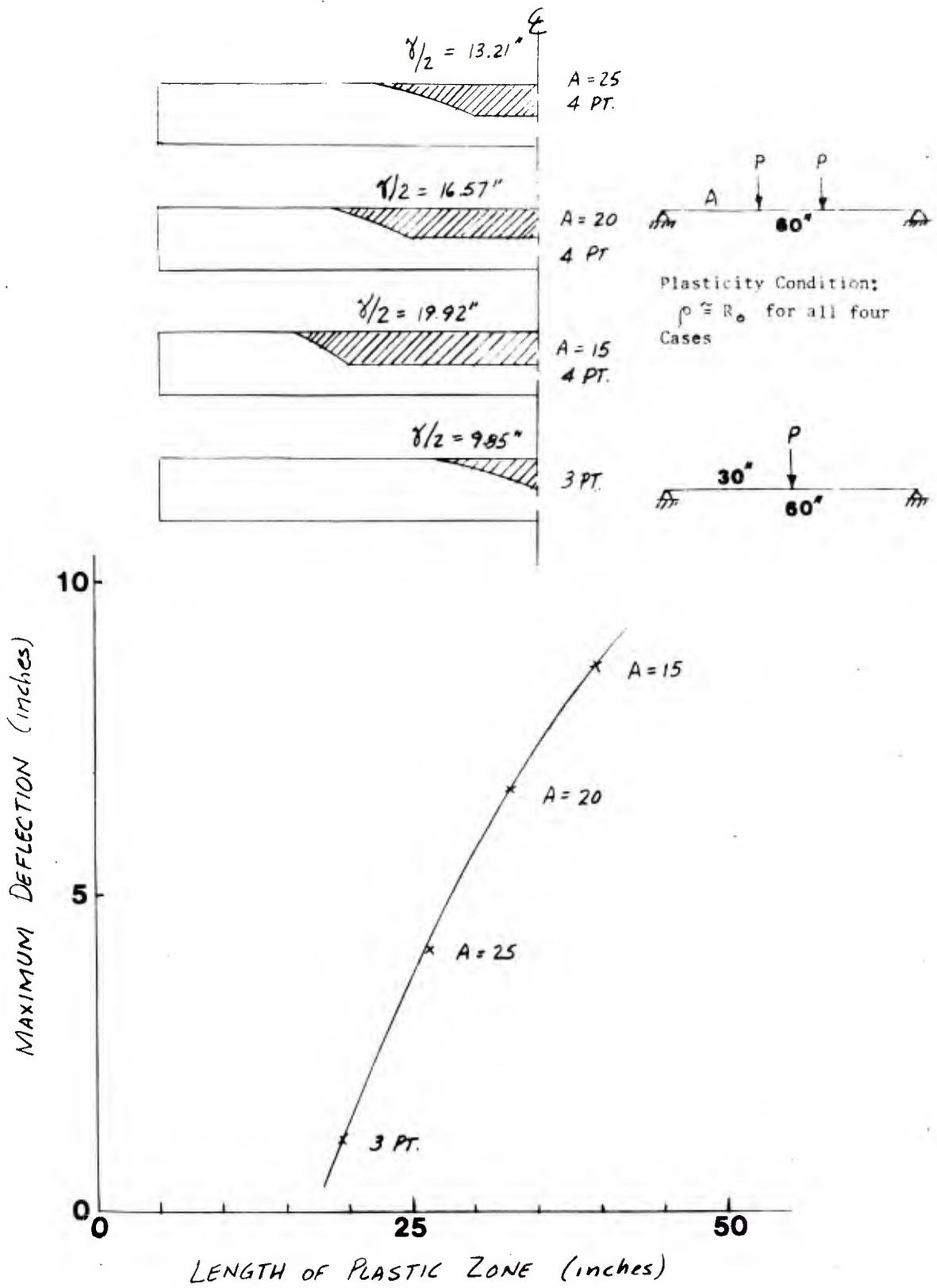


Figure 8. Maximum Deflection Under Load vs. Length of Plastic Zone.

## APPENDIX A

In a previous report<sup>2</sup> the bending moment corresponding to an elastic-plastic interface depth  $\rho$  for a rectangular beam was developed and shown to be

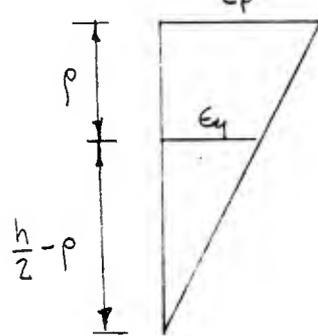
$$M_\rho = \frac{\sigma_y b}{6} [h^2 + 2\rho h - 2\rho^2] \quad (A1)$$


Figure A1. Strain vs. Depth of Elastic-Plastic Interface.

From Figure A1 we get the following relationship

$$\frac{\epsilon_\rho}{\epsilon_y} = \frac{1}{1 - \frac{2\rho}{h}} \quad (A2)$$

Solving for  $\rho$ , we obtain

$$\rho = \frac{h}{2} \left( 1 - \frac{\epsilon_y}{\epsilon_\rho} \right) \quad (A3)$$

Now substituting this into equation (A1), we get the bending moment corresponding to the elastic-plastic interface being at a depth equal to  $\rho$ .

$$M_\rho = \frac{\sigma_y b h^2}{6} \left[ \frac{3}{2} - \frac{1}{2} \frac{\epsilon_y^2}{\epsilon_\rho^2} \right] \quad (A4)$$

---

<sup>2</sup>R. V. Milligan, "Moment-Strain Relationships in Elastic-Plastic Bending of Beams," to be published.

Dividing this expression by the yield moment  $M_y = bh^2/6$  we get the moment ratio

$$\frac{M_p}{M_y} = \frac{3}{2} - \frac{1}{2} \frac{\epsilon_y^2}{\epsilon_p^2} \quad (A5)$$

Multiplying and dividing the second term of equation (A5) by  $(dx/\frac{h}{2})^2$  we get

$$\frac{M_p}{M_y} = \frac{3}{2} - \frac{1}{2} \frac{(\frac{\epsilon_y dx}{h/2})^2}{(\frac{\epsilon_p dx}{h/2})^2} \quad (A6)$$

Now using the results of equations (A4) and (A5) we have

$$\frac{M_p}{M_y} = \frac{3}{2} - \frac{1}{2} \frac{(d\alpha_y)^2}{(d\alpha)^2} \quad (A7)$$

With the view of attempting to make equation (A7) more general so that it can be used for beam cross sections other than rectangular, Seely and Smith substitute  $K$  for  $3/2$  and  $n = K - 1$  for  $1/2$ .  $K$  now represents a geometrical shape factor equal to  $M_{fp}/M_y$ , i.e., the ratio of the fully plastic moment to the yield moment. Table 6.1 of Faupel's book<sup>3</sup> gives values of  $K$  for different beam cross sections. Substituting these parameters, we have

$$\frac{M_p}{M_y} = K - n \left( \frac{d\alpha_y}{d\alpha} \right)^2 \quad (A8)$$

---

<sup>3</sup>J. H. Faupel, Engineering Design, Wiley and Sons, 1964, p. 395.

Solving for  $d\alpha$

$$K - \frac{M_p}{M_y} = n \left( \frac{d\alpha_y}{d\alpha} \right)^2$$
$$d\alpha^2 = n \frac{d\alpha_y^2}{K - \frac{M_p}{M_y}}$$
$$d\alpha = \frac{n^{1/2} d\alpha_y}{\sqrt{K - \frac{M_p}{M_y}}} \quad (A9)$$

Using Hooke's law and the flexure formula and substituting for  $\epsilon$  in (B4) we have

$$d\alpha_y = \frac{M_y}{EI} dx$$

Substituting this into equation (A9) we obtain the desired expression:

$$d\alpha = \frac{M_y n^{1/2} dx}{EI \sqrt{K - \frac{M_p}{M_y}}} \quad (A10)$$

## APPENDIX B

At incipient yielding on the outside fiber we can determine an expression for  $d\alpha$  as follows. By referring to Figure B1 which indicates that planes remain plane and further assuming that the tangent of the angle equals the angle for small deformations, we have

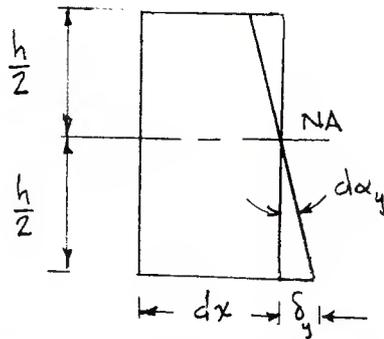


Figure B1. Strain vs. Depth of Cross-Section.

$$\tan(d\alpha_y) = d\alpha_y = \frac{\delta_y}{h/2} \quad (B1)$$

From an expression for engineering strain, we obtain

$$\epsilon_y = \frac{(dx + \delta_y) - dx}{dx} = \frac{\delta_y}{dx} \quad (B2)$$

hence

substituting into equation (A1), we get

$$d\alpha_y = \frac{\epsilon dx}{h/2} \quad (B5)$$

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