Title

Stress Analysis of Large Diameter Taper Hub Pipe Flanges

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STRESS ANALYSIS OF LARGE DIAMETER TAPER HUB PIPE FLANGES

by

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ABSTRACT

Methods of stress analysis for taper hub flanges are investigated, with particular emphasis on flanges for large diameter pipelines. Finite element analyses using (1) an axisymmetric solid idealization and (2) a thin shell idealization are carried out, and compared with each other and with experimental results. It is concluded that the thin shell idealization is most suitable for practical analysis, especially when the inability of the gasket to resist tension is taken into account.

A computer program for practical flange analysis using the thin shell idealization is described.

1. INTRODUCTION

1.1 HISTORICAL REVIEW

Interest in the development of a rational procedure for the design of flanges dates back more than fifty years. One of the earliest papers to receive wide attention was that by Waters and Taylor in 1927 [1]. This paper presented an extensive series of formulas based on a combination of flat plate and elastically supported beam theories. It appears to be the first time that stresses in the three principal directions (hoop, radial and axial) were considered, with the aim of determining the location and magnitude of the maximum stress. The paper also included formulas for computation of the deflection of the flange ring. The computed deflections were compared with experimental results, which were also reported. Because of the proportions of the flanges used at that time, the hoop stress in the ring on the inner surface was critical.

A subsequent important contribution was an analysis procedure for flanges with both rings and tapered hubs, presented by Timoshenko. Most of the formulas which he developed appear in his text book on strength of materials [2].

In 1931, Holmberg and Axelson [3] presented a paper containing a series of formulas for the computation of stresses in loose ring flanges and in flanges made integral with the wall of a pressure vessel or pipe, using flat plate theory. In 1937, Jasper, Gregersen and Zoellner [4] presented the results of an extensive series of tests on plaster-of-paris models. Also in 1937, a series of formulas was presented by Waters, Wesstrom, Rossheim and Williams [5]. The formulas applied to taper hub flanges under bolt load and internal pressure. The structure was subdivided into three components, namely the ring, hub and shell. Each component was studied as an independent unit with undetermined boundary conditions at the junction surfaces. The Poisson - Kirchoff theory for plates was applied to the ring, the hub was treated as a beam of varying section on an elastic foundation, and shell theory was applied to the pipe extension. The boundary conditions were determined by satisfying continuity conditions at the component boundaries. Probably for the first time, the axial symmetry of the flange was taken into account, which considerably simplified the analysis. Displacements, bending moments and stresses in the hub and shell were computed for bolt load and internal pressure, separately and in combination. The paper did not deal rationally with the gasket, or with the contact pressures required to maintain pressure tightness.

A paper dealing with the loading requirements for gaskets in bolted joints was published in 1949 by Roberts [6]. He derived equations to predict the gasket and bolt stresses resulting from application of internal fluid pressure, and presented typical elastic recovery curves for an asbestos gasket. The effect of gasket creep in a bolted joint and the problem of distribution of bolt load were also considered, and an approximate theory derived.

In 1943 recommendations were made by a joint API-ASME committee [7], using an approach similar to that of Waters et al [5]. The results of analyses indicated that in a flange with a straight hub the following stresses tend to be critical:

(a) radial stress at the inside surface of the ring;

(b) hoop stress at the inside surface of the ring;

(c) axial stress in the hub at the junction with the ring.

In 1951 a simplified design procedure using the ASME flange stress formulas was presented by Jaep [8]. The paper contained graphs to permit a rapid solution of stress equations. However the important assumption was made that the bolt load and lever arm are predetermined.

A further step towards the development of a rational procedure was made by Wesstrom and Bergh [9] in 1951. This appears to be the first time that the importance of the bolt load variation and gasket properties were recognized, and an attempt was made to take them into account. It was shown that the bolt stress could either increase or decrease with increasing pressure, depending on the elastic characteristics of the joint. To permit comparison of calculated and experimental results, a bolted flange joint was investigated. The authors obtained fairly good agreement between the calculated and experimental results. Although the authors were unable to propose a rational means of choosing a suitable value for the elastic modulus for the gasket, they recognized the fact that the stress-strain relationships for gasket materials are nonlinear.

Some interesting conclusions on the behavior of compressed asbestos fiber gaskets in bolted flanged joints were reached in 1957 by Donald and Salomon [10], who carried out experiments on screwed-on and welded-on flanges fitted to 2-1/2 inch bore pipes. Rubber-bonded asbestos gaskets of various dimensions were tested, and photoelastic studies were carried out to determine the distribution of axial stress in the gasket. It was shown that for a soft joining material, such as compressed asbestos fiber, the Young's modulus and compressibility depend to a large extent on the stress in the gasket material, and to a lesser extent on its thickness. The location of the gasket reaction depended on the relative stiffness of the flange and the gasket material. A change in the elastic properties of the gasket material, brought about by an increase in the load on the gasket, was found to affect the lever arm between the bolt load and gasket reaction.

Murray and Stuart in 1962 [11] made use of a computer to carry out analyses of large taper hub flanges using a similar approach to that of Waters et al [5]. Two assumptions were that the three components of the flange (cylinder, flange ring and taper hub) have a common diameter equal to the mean diameter of the vessel, and that the bolt holes do not affect the stiffness of the flange ring. It was also assumed that the hub was part of a tapered cylinder of mean radius equal to that of the cylindrical part. Stresses, deflections and rotation of the flange were computed. A rubber O-ring gasket was assumed. The authors also carried out a test on a 15 ft diameter taper hub flange and vessel. The results obtained from the test included (a) longitudinal and hoop stresses on the outer surface of the vessel near the flange (b) the relationship between bolt stress and internal pressure (c) the magnitude of the interface load, and (d) the relationship between flange rotation and pressure. Comparisons indicated that the theoretical analysis was fairly accurate.

Hamada, Ukaji and Havashi [12] also described a computer program for the analysis of stress and deformation in bolted flanges. As is common in other methods, the flange was idealized as three components, namely shell, hub and ring. The shell could be either cylindrical or spherical, and the hub was treated as a cylindrical shell of variable thickness. The ring was considered using circular ring theory, and beam theory was applied to the bolts. The analysis was made on the assumption that the bolt loads were distributed uniformly in the circumferential direction. The program considered loss of rigidity due to the presence of bolt holes in the flange ring. Flange rotation, bolt stresses and bending stresses were computed. The bolt tightening forces and internal pressure loadings were considered. The gasket assumed in the analysis was of the self energizing type, such as an O-ring. The computer analysis compared fairly well with experimental results obtained by the authors, and with those given by Murray and Stuart [11].

Campen, Deen and Latzko [13] reported analytical and experimental work on the deformation of large diameter pressure vessel flanges. Attention was focussed on the question of whether the tapered hub should be treated as a ring with undeformable cross-section or as a thin cylindrical shell of variable thickness. It was concluded that both methods are in sufficiently close agreement with the measurements for design purposes. This paper also provided information on the plastic behavior of the metal-to-metal contact faces and the resulting shift in the point of application of the reaction.

One of the most widely used methods of flange design is the Taylor Forge method [14] pioneered by the Taylor Forge and Pipe Works, Chicago. This method is incorporated in the A.S.M.E. Code. The method contains suggested values for the initial seating load and the load required to seal at pressure for a comprehensive range of gasket types. No account is taken of change in the bolt load on application of pressure, and in fact no direct assessment of the load-deformation characteristics of the joint is made. The formulas and charts used for calculating stresses are based on the work of Waters et al [5]. As is commonly done, the structure is idealized as three components, namely shell, hub and ring. The method is generally found to give lower stresses than the method proposed by Murray and Stuart [11].

A review of methods in current use has been presented by Rose [15], one of which is the European Din 2505 method [16]. This method has been in use in the German chemical industry for several years. It has been tested on flanges up to 2m diameter, and satisfactory service with large diameters is claimed. In addition to values for the initial seating force and the sealing force at pressure when the contained fluid is a liquid or a gas, the code gives maximum permissible loads under assembly conditions and at elevated temperatures for a range of gasket materials. This information is used to check whether or not the gasket is crushed in a vessel which remains hot after pressure is released. This code also gives values for the modulus of elasticity of non-metallic gaskets at 20°C and 300°C.

The flange is designed on the basis of the plastic collapse moment for the flange and shell assembly. Formulas are developed for calculating the collapse moment for a plate type flange. It is assumed that the ring and hub are both fully plastic, the former under hoop stresses and the latter under longitudinal bending stresses. The plastic collapse moment is determined as the sum of the resistances of these two components. Din 2505 uses a method of flange analysis which appears to be relatively simple compared with that of the Taylor Forge method, but which employs a sophisticated approach to the question of leak-tightness. However, this method is too different from accepted U.S. practice to be considered in this report.

1.2 METHODS USED IN PRESENT STUDY

Two methods of analysis have been used in the present study, both of which assume elastic behavior and axisymmetric geometry.

In the first method (thin shell analysis) the common assumption is made that the system can be idealized as a combination of (1) a ring of rigid cross section, representing the flange, (2) a thin conical shell of varying thickness, representing the hub, and (3) a thin cylindrical shell of constant thickness, representing a length of pipe welded to the hub. Although this same idealization has been used by several previous workers, the study reported here differs in the following ways from earlier work.

(a) The conical and cylindrical shells are analyzed by the finite element method, rather than by classical closed form procedures.

(b) The nonlinear characteristics of the gasket material are considered. In particular, the inability of the gasket to develop tension stresses is taken into account.

(c) Changes in bolt force during loading are determined, and predictions can be made of the pressure at which leakage past the gasket will occur.

The method determines hoop and longitudinal stresses on the inner and outer surfaces of the hub and pipe extension, hoop stresses in the flange ring, bolt forces, gasket stresses, and flange rotations. Loadings due to bolt tensioning, internal pressure and axial load can be considered, but bending moment in the pipe is not included.

In the second method (axisymmetric solid analysis) the system is idealized as an axisymmetric solid, so that the cross section of the flange ring need not be assumed to be rigid. The finite element method is used to analyze the solid. Loadings due to bending moment in the pipe, as well as bolt tensioning, internal pressure and axial force, can be considered. However, the analysis is entirely linear, and the inability of the gasket springs to resist tension is not taken into account. The procedure is also more complex than that of the thin shell analysis, and hence is not recommended for use in practical design. The axisymmetric solid analysis has been used in this study to provide a check on the thin shell analysis, and to investigate the effects of bending moment on the flange behavior.

In Chapter 2 these two methods are applied in a series of studies to compare theoretical and experimental results, with particular emphasis on 48 inch taper hub flanges. It is concluded in Chapter 3 that the thin shell idealization is suitable for practical analysis of flanges. The features and use of a computer program based on this idealization are described in Appendix A.

2. THEORETICAL AND EXPERIMENTAL COMPARISONS

2.1 INTRODUCTION

Several analyses have been carried out in the current study to compare theoretical results obtained with the two alternative methods of idealization, and to compare these theoretical results with experimental values obtained from tests on flange joints. The investigations have been as follows:

(1) A comparison of an axisymmetric solid analysis with experimental results reported by Hamada et al [12]. This comparison serves to verify the accuracy of the axisymmetric solid idealization.

(2) A comparison of analyses with axisymmetric solid and thin shell idealizations for a 48 inch diameter pipe flange, assuming linear behavior for the gasket material. This comparison serves to check that the two methods of idealization give similar results, and hence to justify the use of the thin shell idealization. Comparisons with experimental results obtained from full scale flange tests carried out at the University of California, Berkeley are also made. However, the gasket properties assumed for these analyses did not take into account separation of the gasket in tension, and hence the theoretical and experimental results can be expected to disagree significantly.

(3) A comparison of the Berkeley test results with analyses using a thin shell idealization and assuming a gasket material with non-linear properties. This comparison serves to justify the use of the thin shell idealization for practical analysis of flanges.

(4) A comparison of strains computed for moment loading with strains computed for an "equivalent" axial force. The analysis was

carried out using the axisymmetric solid idealization and assuming a linear gasket material. The comparison serves to demonstrate that it is reasonable to make use of an equivalent axial force to approximate the effects of bending moment on a flange joint.

(5) A study of the behavior of a 48 inch flange joint for increasing values of internal pressure up to the stage where leakage past the gasket is predicted. This study was made using the thin shell idealization and nonlinear gasket properties. The study serves to illustrate how gasket stresses and bolt forces vary nonlinearly as the pressure load is increased, and demonstrates the capabilities of the computer program developed for the thin shell idealization.

2.2 CHECK OF AXISYMMETRIC SOLID IDEALIZATION

For linear elastic behavior of both the flange and the gasket, the axisymmetric solid idealization should provide the most accurate method of analysis. Tests of a flange with a "rigid" gasket have been reported by Hamada et al [12]. This flange was idealized using a fine finite element mesh of axisymmetric solid elements and analyzed with the computer program SAP [19]. Comparisons of the computed and measured strains on the inner and outer surfaces of the hub and pipe are shown in Fig. 1. The agreement is seen to be sufficiently close to confirm the accuracy of the axisymmetric solid idealization.

2.3 COMPARISON OF AXISYMMETRIC SOLID AND THIN SHELL IDEALIZATIONS

A 48 inch diameter joint of the type tested at Berkeley was modelled using both axisymmetric solid and thin shell idealizations. The finite element subdivisions in both cases were sufficiently fine to ensure that discretization errors were small. The axisymmetric solid model was analyzed using the program SAP, and the thin shell model using the program AXIFLAN (see Appendix A). The geometry assumed for the flange can be determined from the AXIFLAN input data listed in Appendix B.

For the actual flange, the stiffness of the asbestos gasket was estimated to be 1650 ksi/inch in compression. However, for this study it was necessary to assume that the gasket can resist both tension and compression, whereas a real gasket can obviously not resist tension. From preliminary analyses, it was found that a reasonable comparison between computed and measured strains could be obtained by assuming a linear elastic gasket (i.e. one able to resist tension as well as compression) with a reduced stiffness of 825 ksi/inch. Hence, such a gasket was assumed for the study. It should be noted that the AXIFLAN program as described in Appendix A can not accept gaskets which resist tension. It was necessary, therefore, to make a minor modification to the program to permit this study to be carried out.

For a bolt pretension of 139 k per bolt, with no other loading, the computed strains are compared in Figs. 2 and 3 for the inner and outer surfaces of the hub and pipe. Measured strains obtained in the Berkeley tests are also shown. In these tests, strain gages were placed at several locations around the flange circumference, so that several strain readings were obtained at each longitudinal section. The plotted points show the maximum ranges in experimental results obtained considering the gages at all circumferential locations.

Except for the following areas of disagreement, the computed strains agree well with each other and with the measured strains.

(1) The measured strains clearly show a longitudinal strain concentration effect at the intersection of the hub with the flange ring. The axisymmetric solid idealization accounts for this strain

concentration, but the effect can not be considered in the thin shell idealization. This is a weakness of the thin shell idealization which must be recognized if this strain concentration effect is believed to be important.

(2) The computed hoop strains on the inner surface are substantially larger than the measured strains. This disagreement was found to be present for most other analyses and loading conditions, and the reason for it is not clear. Because the test specimen was filled with water which was frequently under high pressure, the possibility of unreliable gage readings should not be discounted. The discrepancies are such that the computed strains are consistently larger than the measured strains, so that the computed stress values should generally be conservative.

For a bolt pretension of 139 k plus an internal pressure of 1200 psi, similar comparisons of computed and measured strains are shown in Figs. 4 and 5. The results from the two analyses are again in close agreement, except that the strain concentration effect at the intersection of the hub and flange is still present. Again, the computed hoop strains on the inner surface substantially exceed the measured strains, and in addition the computed longitudinal strains on the inner surface are well below the measured values. However, the results of the next study show that the second discrepancy results from the assumption that the gasket can resist tension. The ranges between the maximum and minimum measured strains at each longitudinal section are seen to be large for both the longitudinal and hoop strains on the inner surface, although the ranges are still small on the outer surface. This difference in range between inner and outer surface gages persists for essentially all load cases, and serves to throw doubt on the reliability of the readings of the inner surface gages.

For Figs. 4 and 5 the test specimen was unrestrained longitudinally, so that the test specimen was subjected to a large longitudinal tension force due to fluid pressure. Figs. 6 and 7 show results for the case with a bolt pretension of 139 k, an internal pressure of 1200 psi and a longitudinal compression load of 2060 k. This loading simulates a pipeline which is fully restrained longitudinally. The axial strains are seen to be considerably reduced, and the agreement among the different results is generally close.

These comparisons serve to demonstrate that with the exception of strain concentration effects of the hub-flange intersection the thin shell idealization produces results in close agreement with the axisymmetric solid idealization, and hence that the thin shell idealization should be suitable for practical analyses. Any disagreement between the computed and measured strains is not of great importance for this part of the study, because the gasket was not accurately modelled.

2.4 THIN SHELL IDEALIZATION WITH NONLINEAR GASKET

The thin shell computer program AXIFLAN permits a nonlinear idealization of the gasket, as described in detail in the program user's guide in Appendix A. Essentially, the gasket is assumed to deform elastically in compression but to be incapable of taking tension. In addition, fluid can be assumed to seep past the gasket if its tensile deformation exceeds a specified "precompression", so that the axial force tending to separate the flanges may progressively increase. The compressive stiffness of the asbestos gasket used in the tests was estimated to be 1650 ksi/inch, with a precompression of 0.015 inches.

Comparisons of the computed and measured strains are shown for a bolt pretension of 139 k in Figs. 8 and 9, for bolt pretension plus 1200 psi internal pressure in Figs. 10 and 11, and for bolt pretension plus pressure plus an axial force of 2060 k in Figs. 12 and 13. The areas of disagreement are as follows:

(1) The strain concentration effect at the hub-flange intersection is not considered in the analysis.

(2) The computed hoop strains generally exceed the measured strains, particularly on the inner surface.

The agreement between the computed and measured longitudinal strains can be seen to be closer in Fig. 10 than was the case in Fig. 4, indicating that the nonlinear gasket assumption is superior to the linear assumption. However, as noted previously there is a large scatter in the measured strains.

It can be concluded from this study that the AXIFLAN program provides a reasonably accurate means of determining stresses and strains in flanges of this type. It should be noted that most other methods of flange stress analysis are also based on thin shell idealizations. Analyses obtained using AXIFLAN can be expected to be at least as accurate as those obtained using these other methods.

2.5 EQUIVALENT AXIAL FORCE

Flanged joints are frequently subjected to bending moment as well as internal pressure and axial force. Because of the difficulty of analyzing the behavior of a flanged joint subjected to bending, it is attractive from a design point of view to replace the bending moment by an equivalent axial force. For a thin walled pipe with a mean radius R and any wall thickness, it is easy to show that an axial force, F, will produce the same longitudinal stress as a bending moment, M, if

$$F = \frac{2M}{R} \tag{2.1}$$

The value of R might be taken as the bolt circle radius or, more conservatively, as the pipe radius. For the 48 inch diameter flange the difference between these two radii is approximately 20 percent.

Using an axisymmetric solid idealization and the SAP program it is possible to analyze the flange under a pure bending moment loading. The loading in this case is not axisymmetric, but is the first harmonic of the Fourier series expansion of an arbitrary loading. A modification was made to the axisymmetric solid element to permit a first harmonic loading to be considered, and an analysis was carried out for a moment of 10550 k.in. The finite element mesh and gasket idealization were the same as those used previously for the axisymmetric solid idealization. In addition, a SAP analysis for an axial force of (2)(10550)/(24) = 879 k was carried out, using the same mesh. The two analyses are compared in Figs. 14 and 15 with each other and with strains measured in the Berkeley tests for a moment loading of 10550 k.in. The strains shown for the bending moment analysis were computed for the fibers most remote from the bending axis, and the measured strains shown are those for gages on or very close to the most remote fibers.

The agreement between the two sets of computed strains is seen to be surprisingly close, indicating that the replacement of a bending moment by an equivalent axial force is permissible. The computed and measured strains also agree closely, except for some longitudinal strains on the inner surface. As was shown previously, this disagreement probably results from the assumed linear gasket idealization. This comparison indicates that R in Eq. 2.1 should be the pipe radius.

2.6 ILLUSTRATIVE EXAMPLE

The AXIFLAN input data for an illustrative example is listed in Appendix B. This example is a 48 inch flange of the type studied in the Berkeley tests. The loading consists of a bolt pretension of 139 k per bolt, followed by pressurization, in steps, until leakage past the gasket is predicted and the computer run automatically terminates. The line was unrestrained longitudinally.

The program predicted leakage past the gasket at a pressure between 1800 and 2000 psi. No experimental data on leakage is available for comparison. The computed bolt tensions and gasket stresses for progressively increasing pressures are shown in Figs. 16 and 17. As is well known, the bolt tension initially decreases, as the flange ring rotates, but later increases, as fluid seeps past the gasket and increases the axial force. The von Mises effective stresses computed by the program are shown for pressures of 0,800 and 1200 psi in Fig. 18. These stresses approach or exceed yield for pressures larger than 1200 psi. Hence, the computed behavior beyond this pressure may be in error because elastic material behavior is assumed in the analysis.

3. CONCLUSIONS

The comparisons presented in Chapter 2 indicate that the strains computed using the AXIFLAN program are in satisfactory agreement with experimental strains, with the major exception that strain concentration effects at the hub-flange intersection can not be predicted. The structural idealization on which AXIFLAN is based is similar to that commonly used in flange design, but is more sophisticated than other analyses which have been reported. In particular, nonlinear behavior of the gasket can be considered directly with AXIFLAN. The program can be used for loads due to bolt tightening, internal pressure, and axial force. The output from the program includes von Mises effective stresses in the hub and pipe. The program includes data generation procedures which make data preparation very simple. It is believed that the program can have application in the practical design of taper hub flanges.

A comparison of stresses produced by a bending moment, M, and an equivalent axial force, $F = 2M/R$, where R = pipe radius, indicates that the moment can be replaced by the equivalent axial force for design purposes.

APPENDIX A: PROGRAM USER'S GUIDE

UNIVERSITY OF CALIFORNIA
Berkeley

Division of Structural Engineering
and Structural Mechanics

Computer Programming Series

IDENTIFICATION

AXIFLAN: Finite Element Thin Shell Analysis of Pipe Flanges

Programmed: G. H. Powell, V. S. Prakash, R. Litton, University of California, Berkeley.

PURPOSE

The program analyzes the behavior of taper hub flanges, with cross sections as shown in Fig. A.1, under the action of bolt tension, internal pressure and axial force. Bolt force, gasket stresses, and stresses on the inner and outer surfaces of the hub and pipe are determined. Leakage past the gasket is taken into account.

IDEALIZATION

The system is idealized as an axisymmetric structure, as indicated in Fig. A.2. The flange ring is treated as a compact ring beam, the hub and pipe are represented by axisymmetric thin shell finite elements, and the gasket and bolt by discrete ring springs. The effect of the bolt holes is considered by reducing the ring beam modulus, within the projected width of the holes, in proportion to the volume of material removed by the holes.

The number of elements into which the hub and pipe are divided is chosen by the user. Experience and experimentation will be needed to select the appropriate numbers. The same is true for the number of discrete springs used to idealize the gasket.

The pipe extension should be sufficiently long that the assumed boundary condition (corresponding to a length of pipe remote from any end disturbances) is reasonable. A length no less than $5\sqrt{Rt}$, where R = pipe radius and t = pipe wall thickness, is recommended.

The gasket springs are assigned the force-deflection relationship shown in Fig. A.3. This approximates the behavior of a gasket material which has been subjected to loading and unloading cycles. Additional pressure loading as shown in Fig. A.4 is applied as the gasket separates.

Loadings are specified in terms of an initial bolt tension, followed by any number of pressure and axial force loadings.

INPUT DATA

Units must be consistent throughout (e.g. kips, inches).

A. TITLE CARD (12A6) - One card

Columns 1 - 72: Problem title, to be printed with output.

B. FLANGE DIMENSIONS (5E10.0) - One card

Columns 1 - 10: Outside diameter of flange (DOFL, see Fig. A.1)

11 - 20: Bolt circle diameter (BCDIA)

21 - 30: Outside diameter of projection (DOPR)

31 - 40: Flange thickness (TFL)

41 - 50: Projection thickness (TPR)

C. HUB AND PIPE DIMENSIONS (5E10.0, 2I5) - One card.

Columns 1 - 10: Outside diameter of pipe (DOPIP)

11 - 20: Pipe wall thickness (TPIPE)

21 - 30: Hub thickness at flange ring (THUB)

31 - 40: Hub length (HUBL)

41 - 50: Length of pipe extension (PIPEL)

51 - 55 Number of finite elements into which hub is to be divided (NEHUB)

56 - 60: Number of finite elements into which pipe extension is to be divided (NEPIP).

Limitation: The sum of NEHUB and NEPIP must not exceed 40.

D. BOLT DIMENSIONS (3E10.0, I5) - One card.

Columns 1 - 10: Diameter of bolt (DBOLT). See Note below.

11 - 20: Effective half length of bolt (BOLTL). See Note below.

21 - 30: Diameter of bolt hole (DHOLE)

31 - 35: Total number of bolts in bolt circle (NBOLT)

Note: The bolts are idealized as a spring at the bolt circle, as indicated in Fig. A.2. The spring stiffness and the resulting bolt forces are calculated assuming each bolt to have a stiffness given by (EBOLT) (π) $(\text{DBOLT})^2/(4)$ (BOLTL). The bolt diameter and length should therefore be effective values, accounting for the effect of threading and for the fact that the bolt extension does not terminate suddenly at the face of the nut. These effects can also be taken into account by specifying an effective elastic modulus for the bolt material in Section E.

E. MATERIAL PROPERTIES (5E10.0) - One card

Columns 1 - 10: Young's modulus for flange ring and hub material (EFLANG).

11 - 20: Poisson's ratio for flange ring and hub material (PRFL)

21 - 30: Young's modulus for pipe material (EPIPE)

31 - 40: Poisson's ratio for pipe material (PRPIP)

41 - 50: Young's modulus (or effective modulus) for bolt material (EBOLT)

F. GASKET PROPERTIES (4E10.0, I5) - One card

Columns 1 - 10: Inside diameter of gasket (DIGASK)

11 - 20: Outside diameter of gasket (DOGASK). This must not be identically equal to DIGASK. For O-ring type gaskets, assume a reasonable gasket width.

21 - 30: Gasket effective stiffness, in terms of stress per unit compression for a gasket of the type and thickness being used, after pre-compression and assuming elastic behavior. See discussion in preamble to this Guide and Note below.

31 - 40: Gasket precompression, as defined in the preamble to this guide, which is the amount of tensile deformation in the gasket before fluid seepage begins at the gasket face. See Note below.

41 - 45: Number of discrete springs to be used to represent gasket (minimum 2, maximum 15). A recommended number is 10.

Note: The stiffness and precompression are for the full gasket thickness. In the analysis, only one half of the thickness is used, because of symmetry. To account for this, the specified stiffness and precompression are multiplied by 2.0 and 0.5, respectively, within the computer program.

G. LOAD CARDS - One set of cards, as follows, for each load sequence, for as many load sequences as desired.

First Card (E10.0, 2I5, 8A6) - One card.

Columns 1 - 10: Tightening force in each bolt.

11 - 15: Number of different pressure and axial force loadings for which analyses are required for this bolt tightening force (NUMLDS).

16 - 20: Leakage termination code. If blank, the loadings in the following sequence will be considered only until complete separation of the gasket, and hence fluid leakage, is predicted. The remaining loadings will be ignored, and the program will proceed to the next load sequence. If not blank, all loadings will be considered, regardless of whether leakage is predicted.

21 - 68: Load sequence title, to be printed with output. The load sequence will automatically be identified by number in the output. Leave title field blank if no additional identification is needed.

Remaining cards of set (2F10.0, 8A6) - NUMLDS cards.

Columns 1 - 10: Internal pressure.

11 - 20: Axial load on pipe, tension positive.

21 - 68: Loading title, to be printed with output. The loading will automatically be identified by number in the output. Leave title field blank if no additional identification is needed.

Note 1: The axial force due to internal pressure, assuming the end of the pipe is closed, is automatically included by the program. The axial load specified in Columns 11 - 20 is the force applied in addition to this pressure force.

Note 2: The program will execute most efficiently if these cards are arranged in increasing values of pressure and/or axial force.

H. NEXT PROBLEM

After all load sequences, insert a blank card. A new problem may then be defined, starting with Card A. To terminate the run, insert two additional blank cards.

APPENDIX B: SAMPLE INPUT DATA

```
EXAMPLE FLANGE.  ALYESKA DIMENSIONS.  UNRESTRAINED LONGITUDINALLY.
  63.314     58.564    54.626     5.063      .25
 48.         .562      2.75       7.063      18.           12    14
  2.5        6.0       2.625       32
30000.      .3        30000.      .3         30000.
 46.876      54.626    1650.      .015        10
 139.         8     0   139K BOLT LOAD. PRESSURE TO LEAKAGE.
 .8
 1.2
 1.6
 1.8
 2.0
 2.2
 2.4
 2.6
```

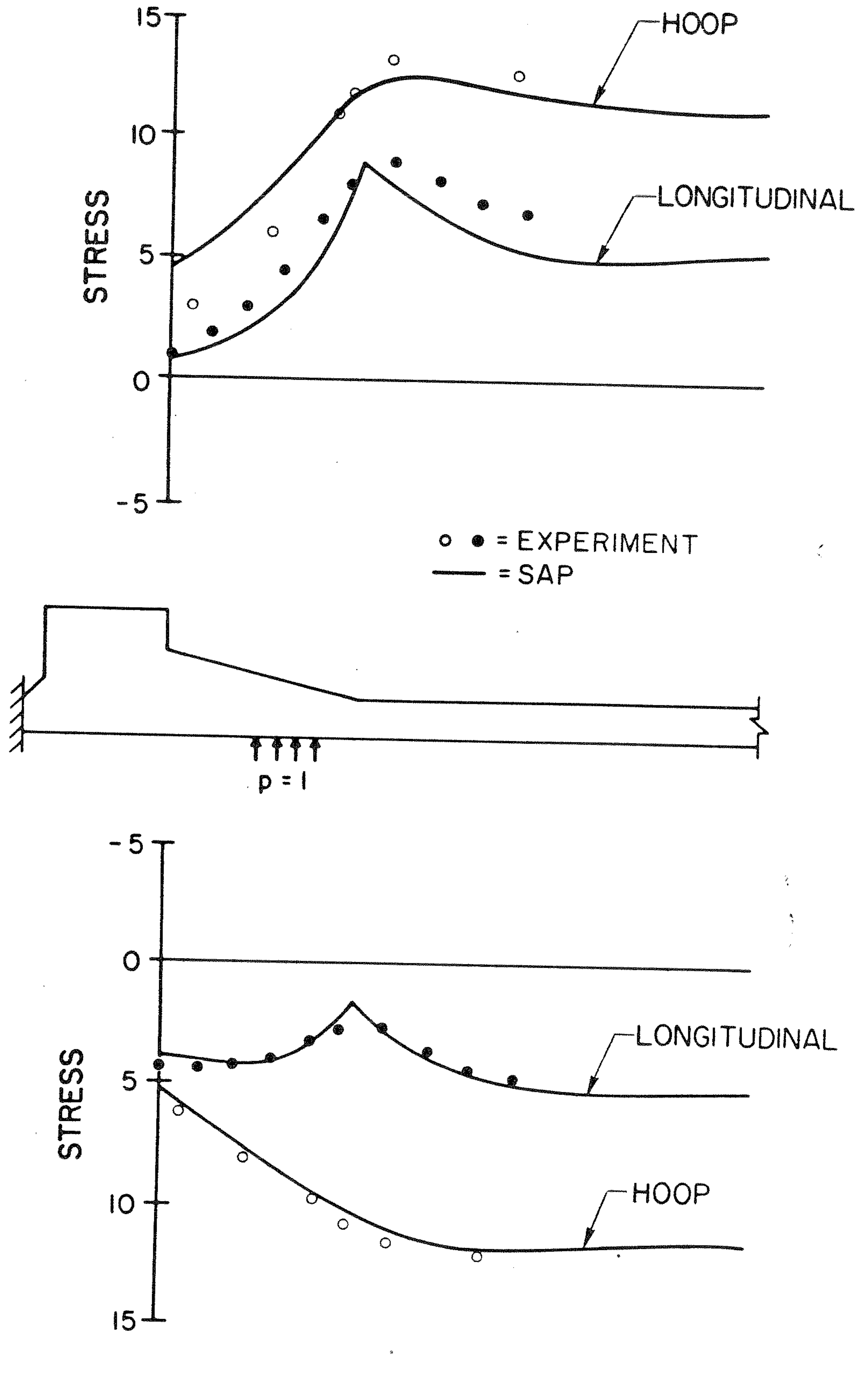

FIG. I SAP ANALYSIS OF FLANGE TESTED BY HAMADA ET AL.

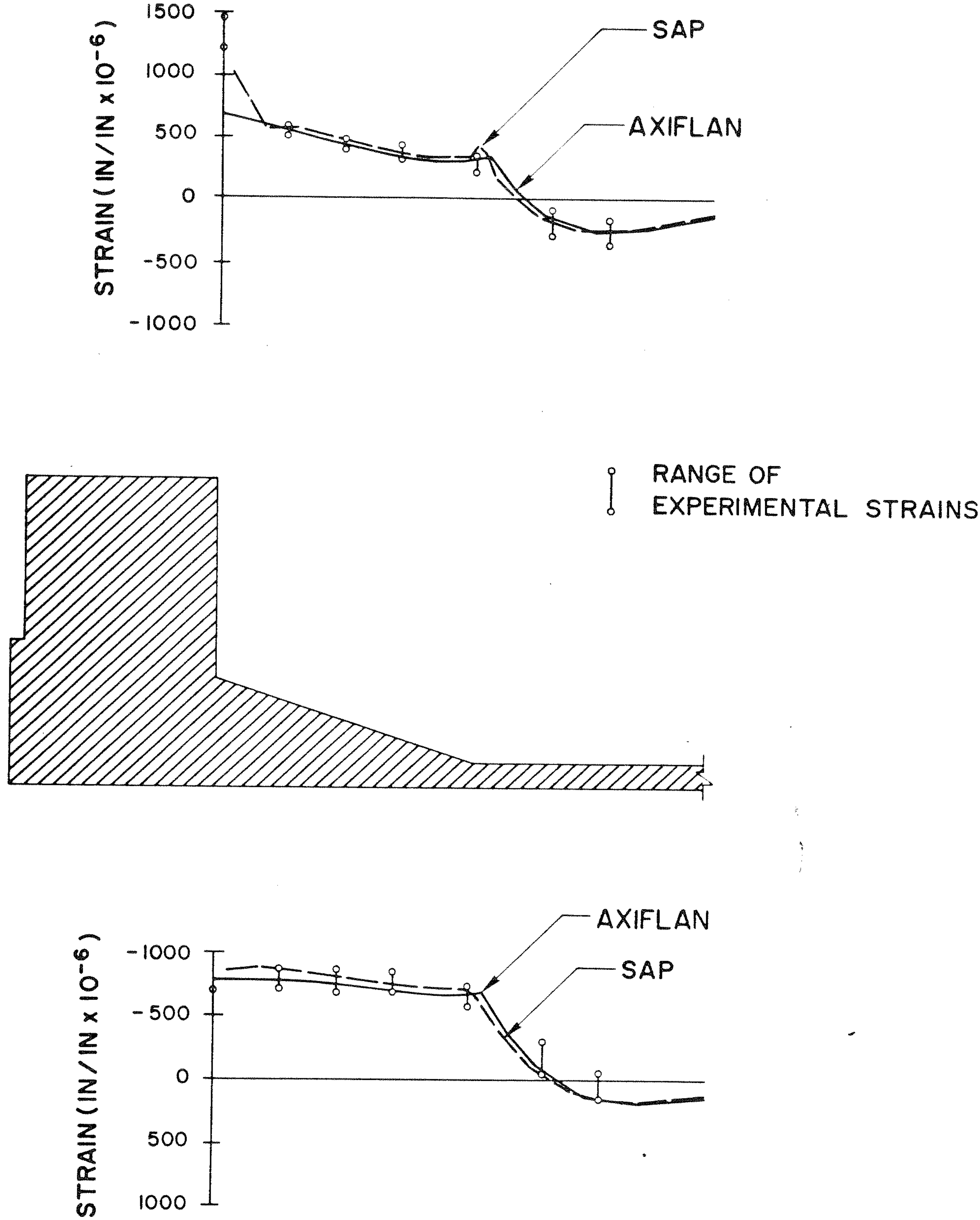

FIG. 2 COMPARISON OF SAP AND AXIFLAN. BOLT LOAD ONLY, LONGITUDINAL STRAIN

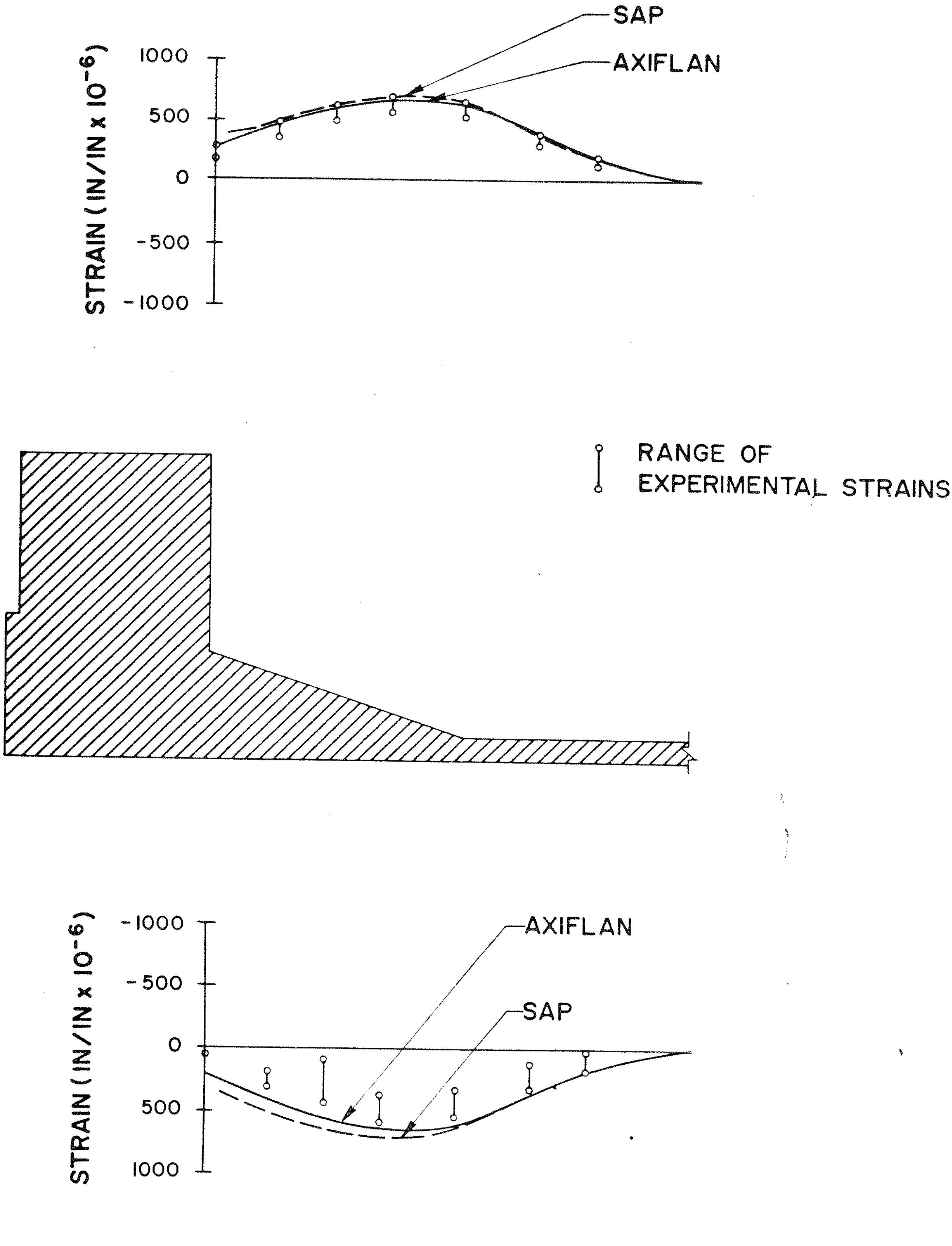

FIG. 3 COMPARISON OF SAP AND AXIFLAN. BOLT LOAD ONLY, HOOP STRAIN.

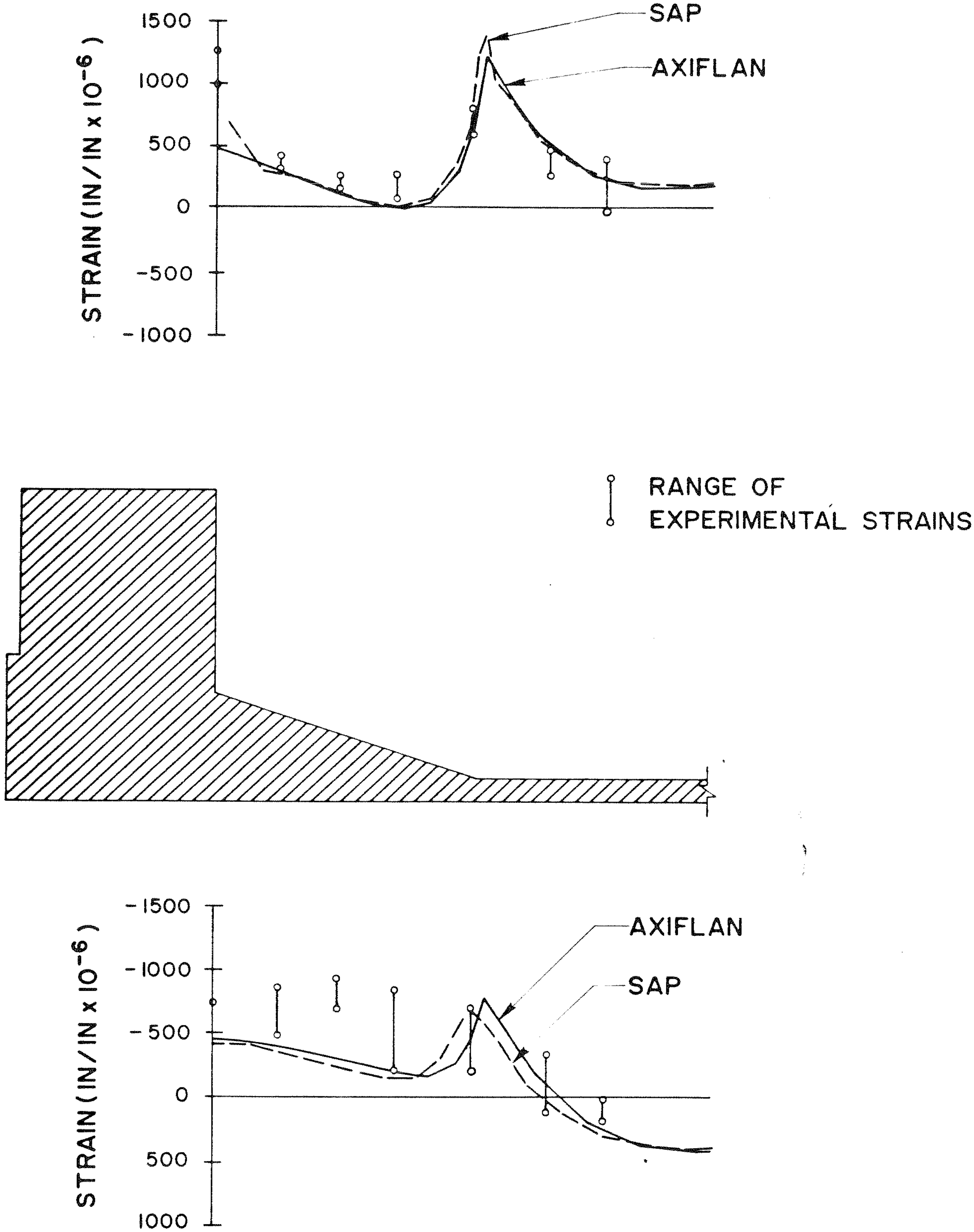

FIG. 4 COMPARISON OF SAP AND AXIFLAN. BOLT LOAD PLUS PRESSURE. LONGITUDINAL STRAIN

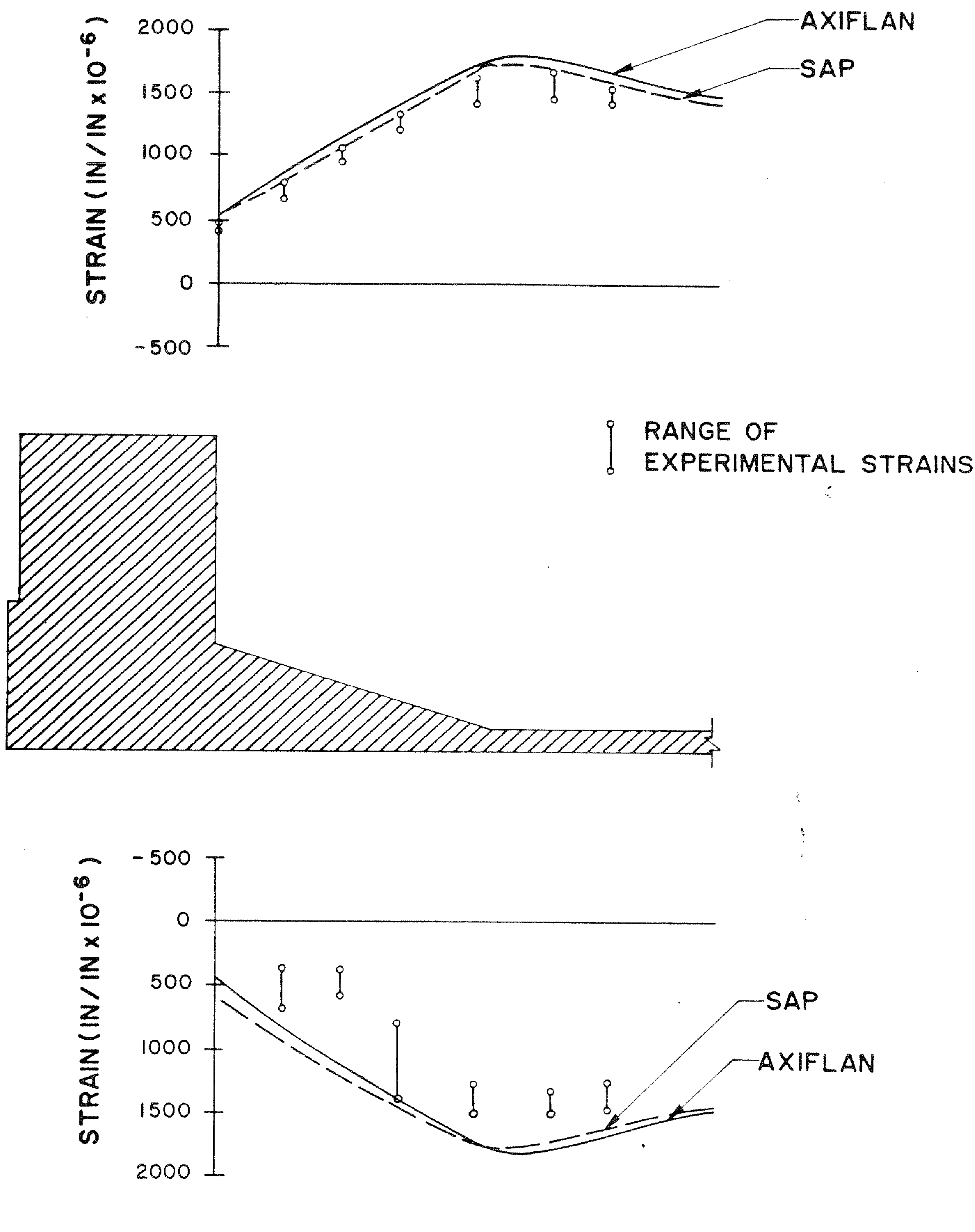

FIG. 5 COMPARISON OF SAP AND AXIFLAN. BOLT LOAD PLUS PRESSURE. HOOP STRAIN.

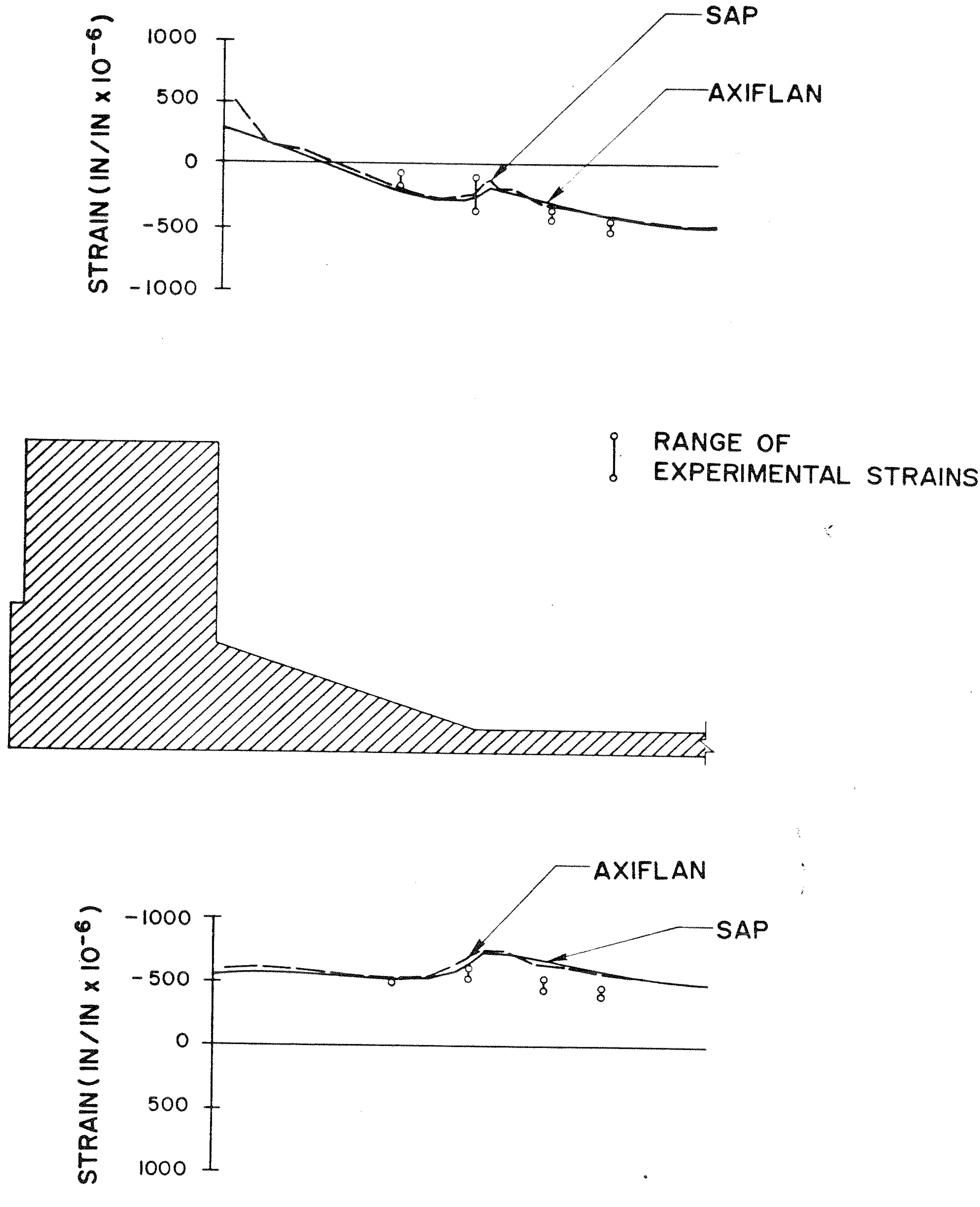

FIG. 6 COMPARISON OF SAP AND AXIFLAN. BOLT LOAD, PRESSURE AND AXIAL FORCE. LONGITUDINAL STRAIN.

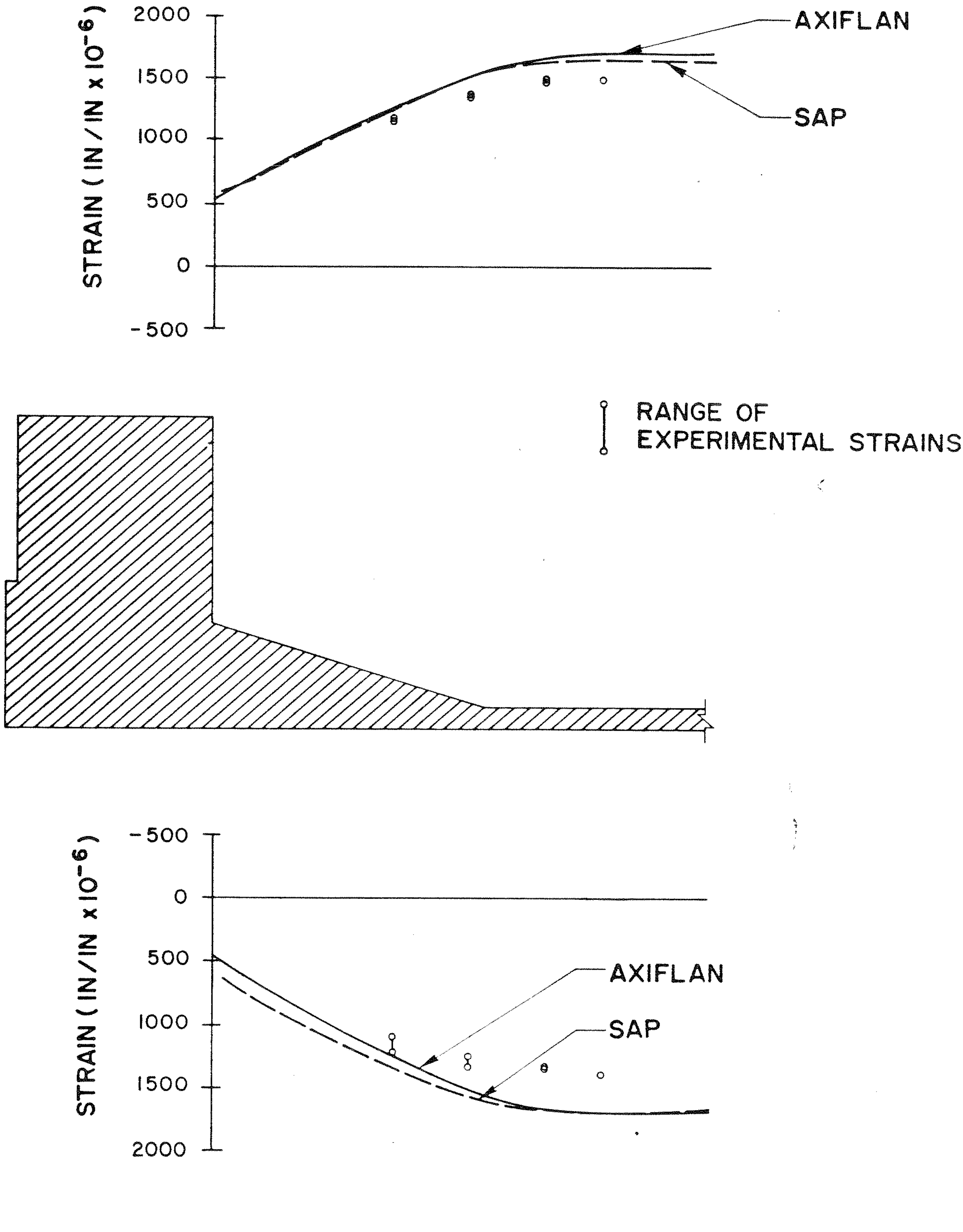

FIG. 7 COMPARISON OF SAP AND AXIFLAN.
BOLT LOAD, PRESSURE AND AXIAL FORCE.
HOOP STRAIN.

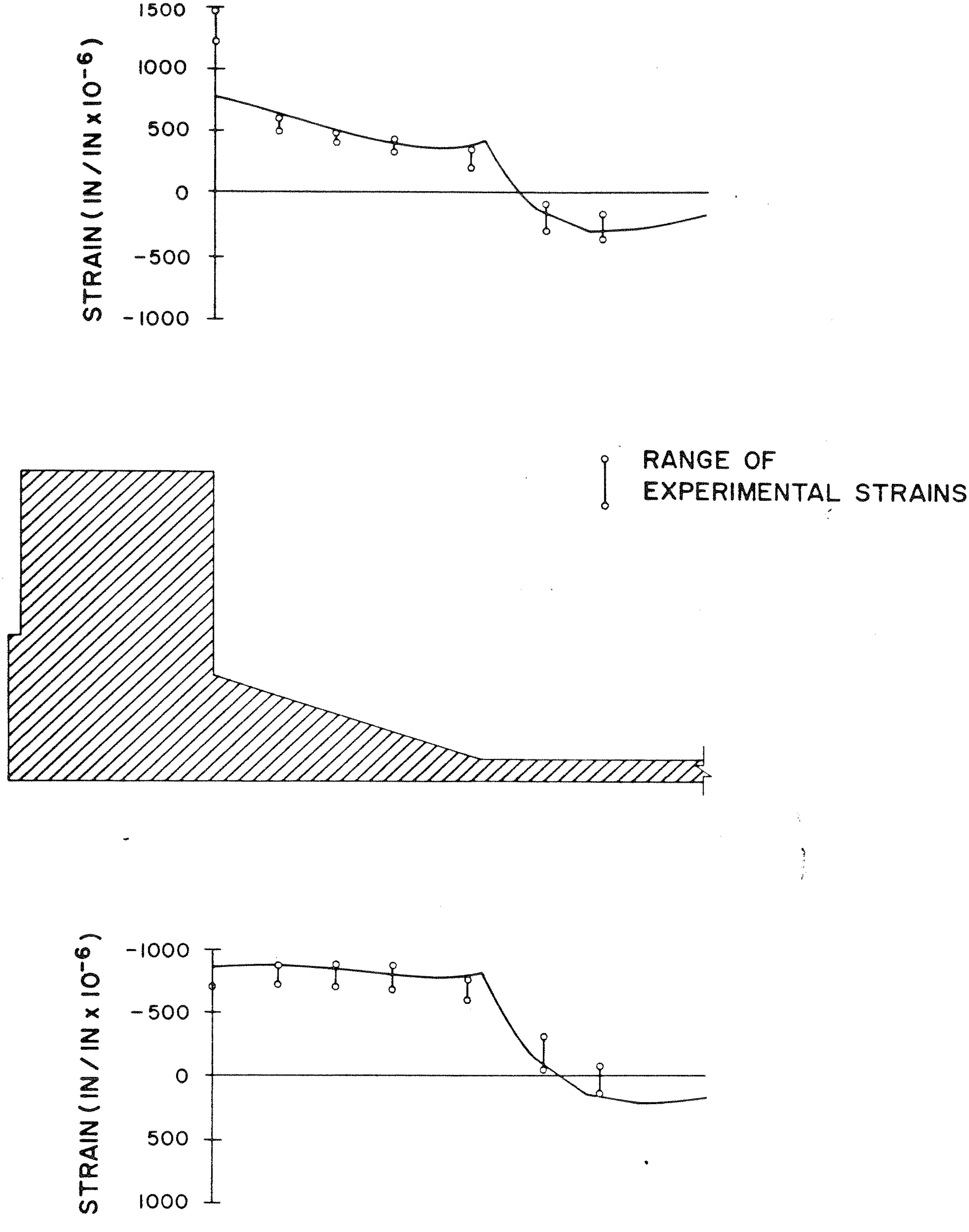

FIG. 8 AXIFLAN ANALYSIS WITH NONLINEAR GASKET. BOLT LOAD ONLY. LONGITUDINAL STRAIN.

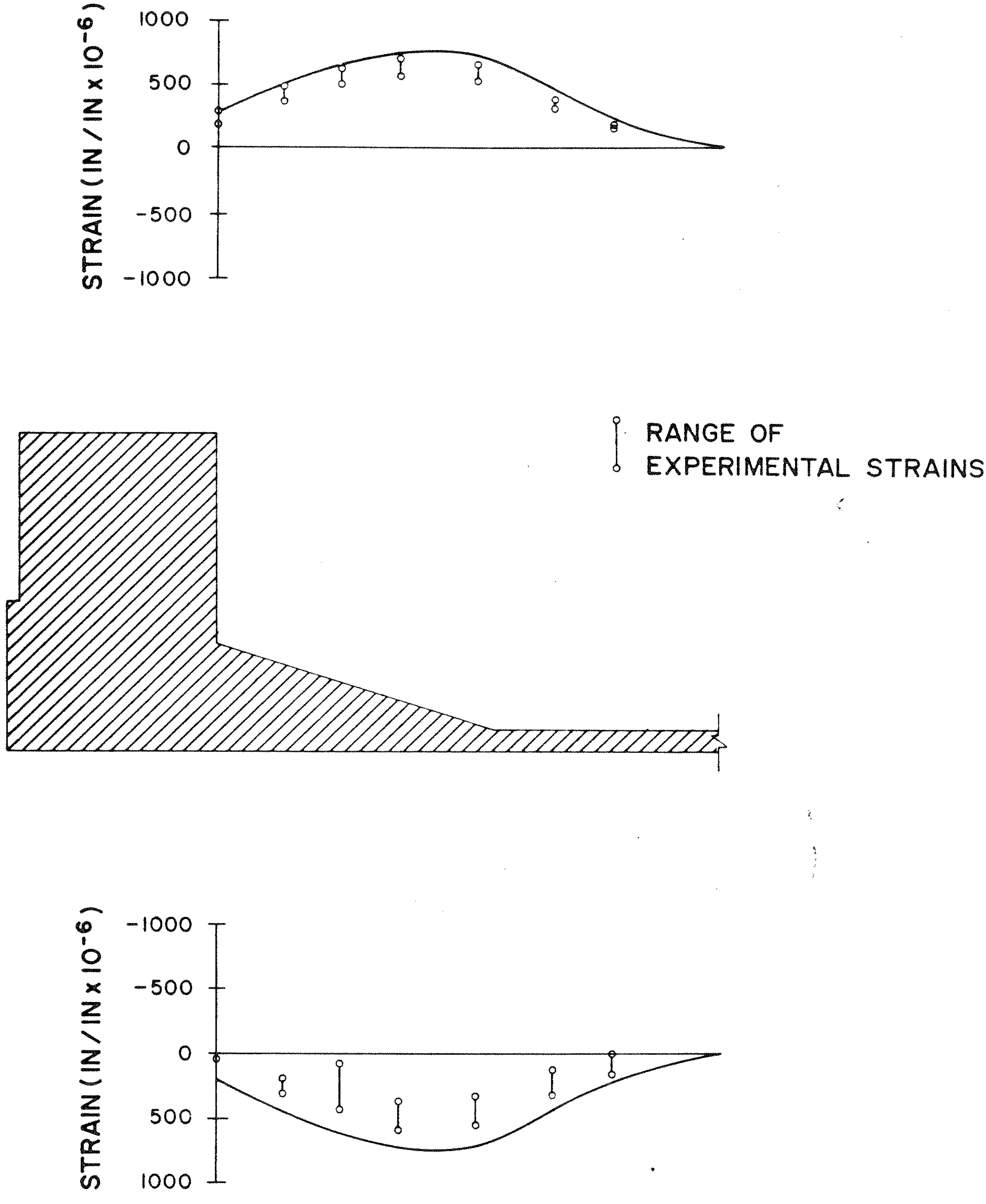

FIG. 9 AXIFLAN ANALYSIS WITH NONLINEAR GASKET. BOLT LOAD ONLY. HOOP STRAIN.

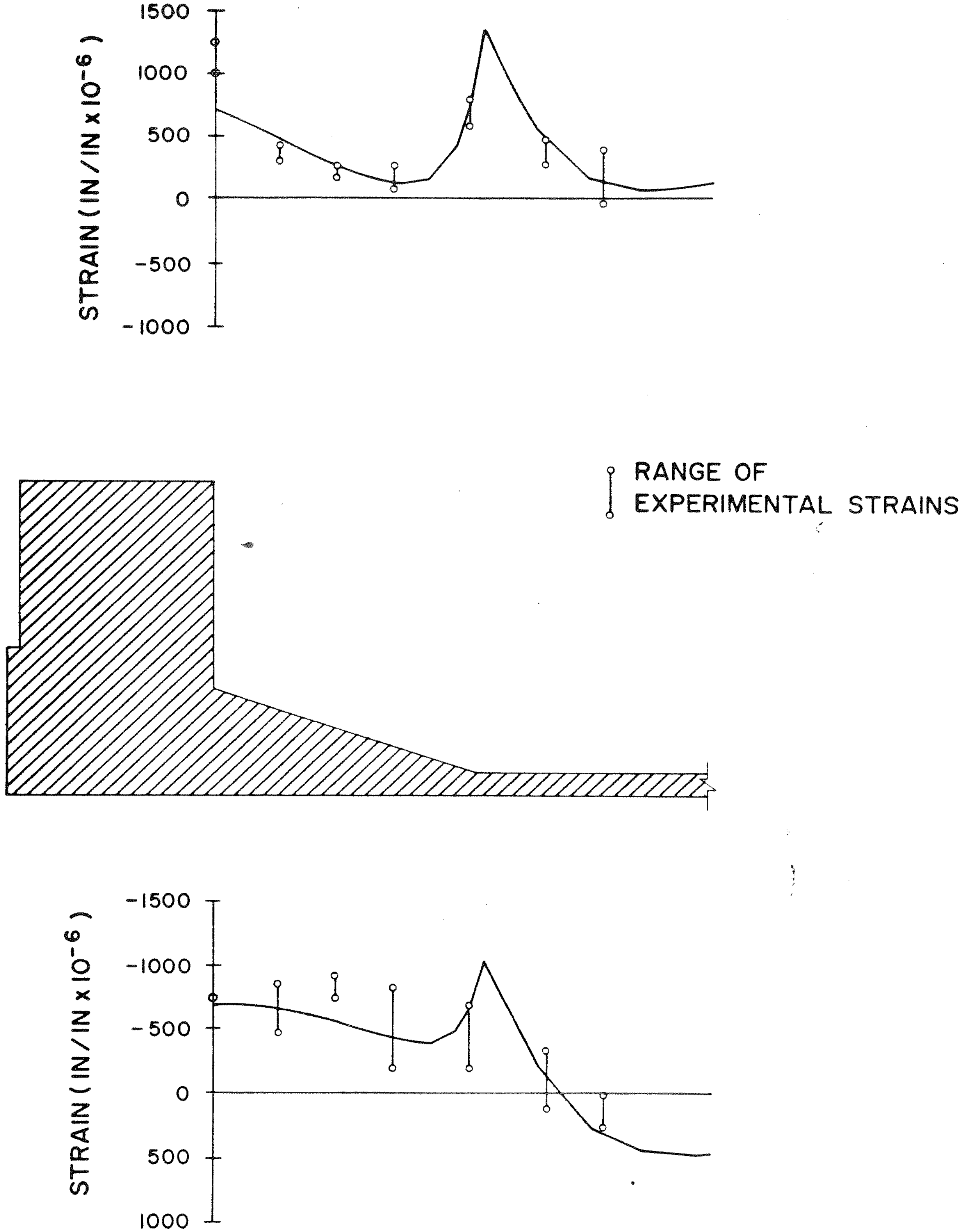

FIG. 10 AXIFLAN ANALYSIS WITH NONLINEAR GASKET. BOLT LOAD PLUS PRESSURE. LONGITUDINAL STRAIN.

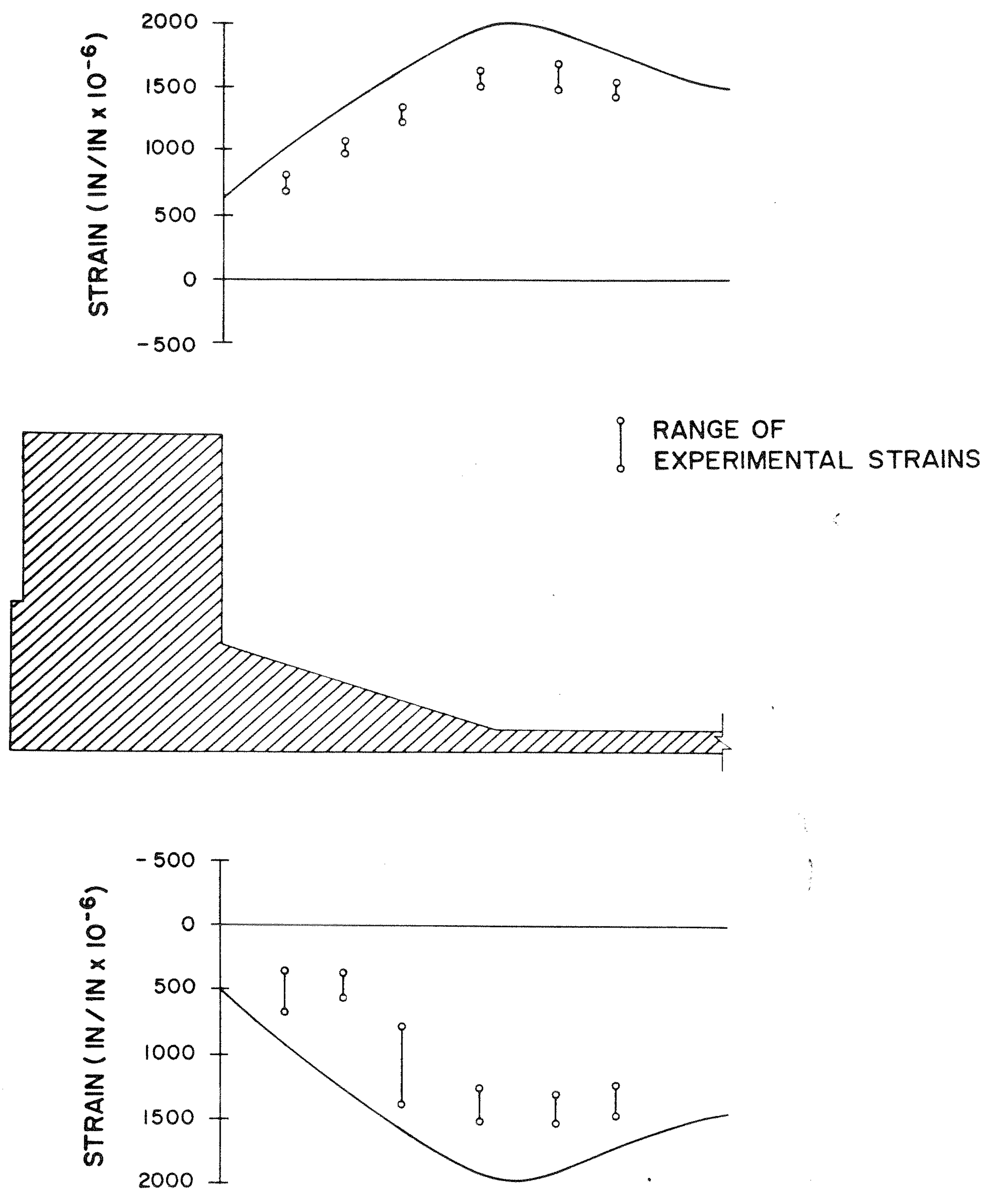

FIG. 11 AXIFLAN ANALYSIS WITH NONLINEAR GASKET. BOLT LOAD PLUS PRESSURE. HOOP STRAIN.

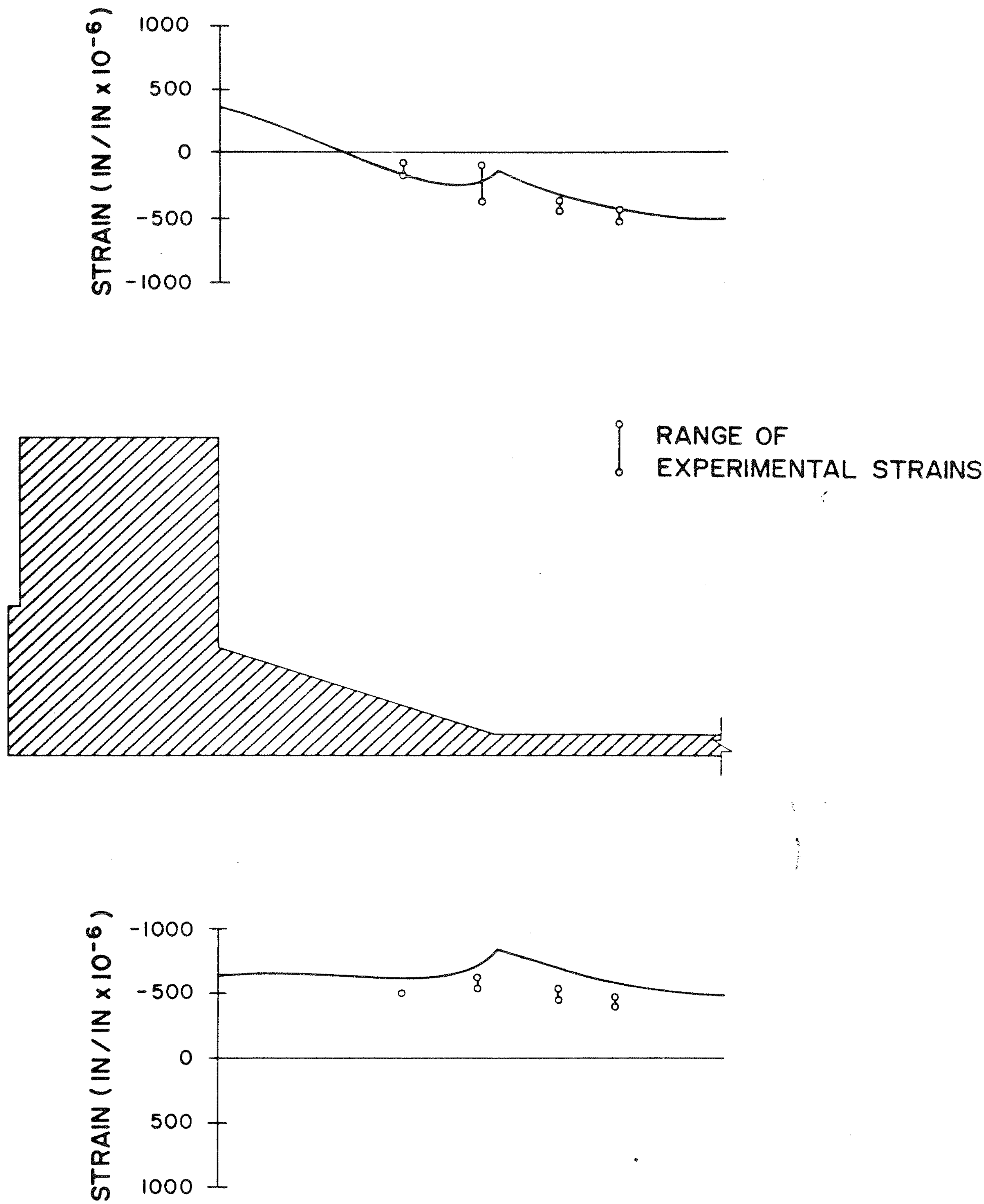

FIG. 12 AXIFLAN ANALYSIS WITH NONLINEAR GASKET. BOLT LOAD, PRESSURE AND AXIAL FORCE. LONGITUDINAL STRAIN.

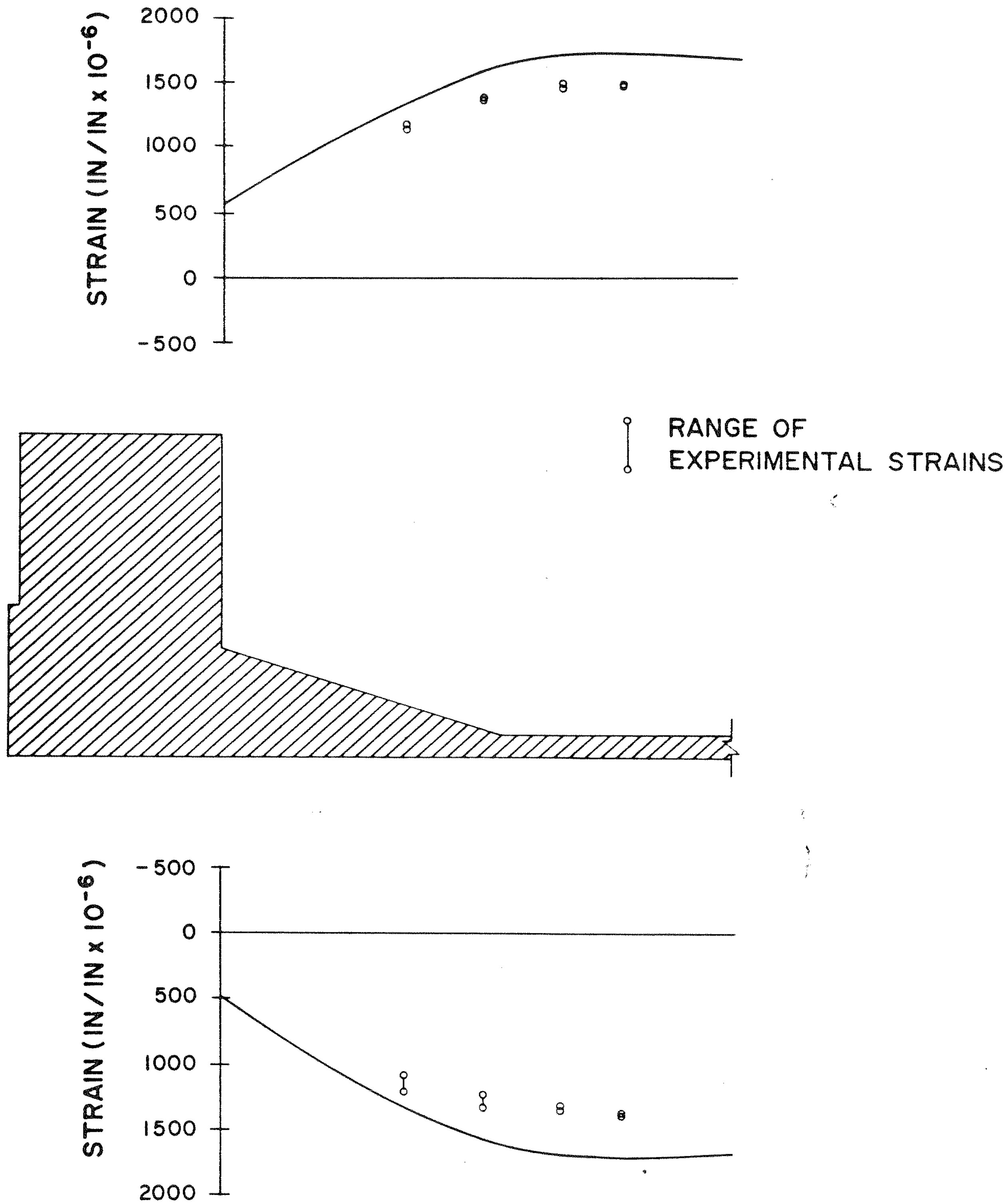

FIG. 13 AXIFLAN ANALYSIS WITH NONLINEAR GASKET. BOLT LOAD, PRESSURE AND AXIAL FORCE. HOOP STRAIN.

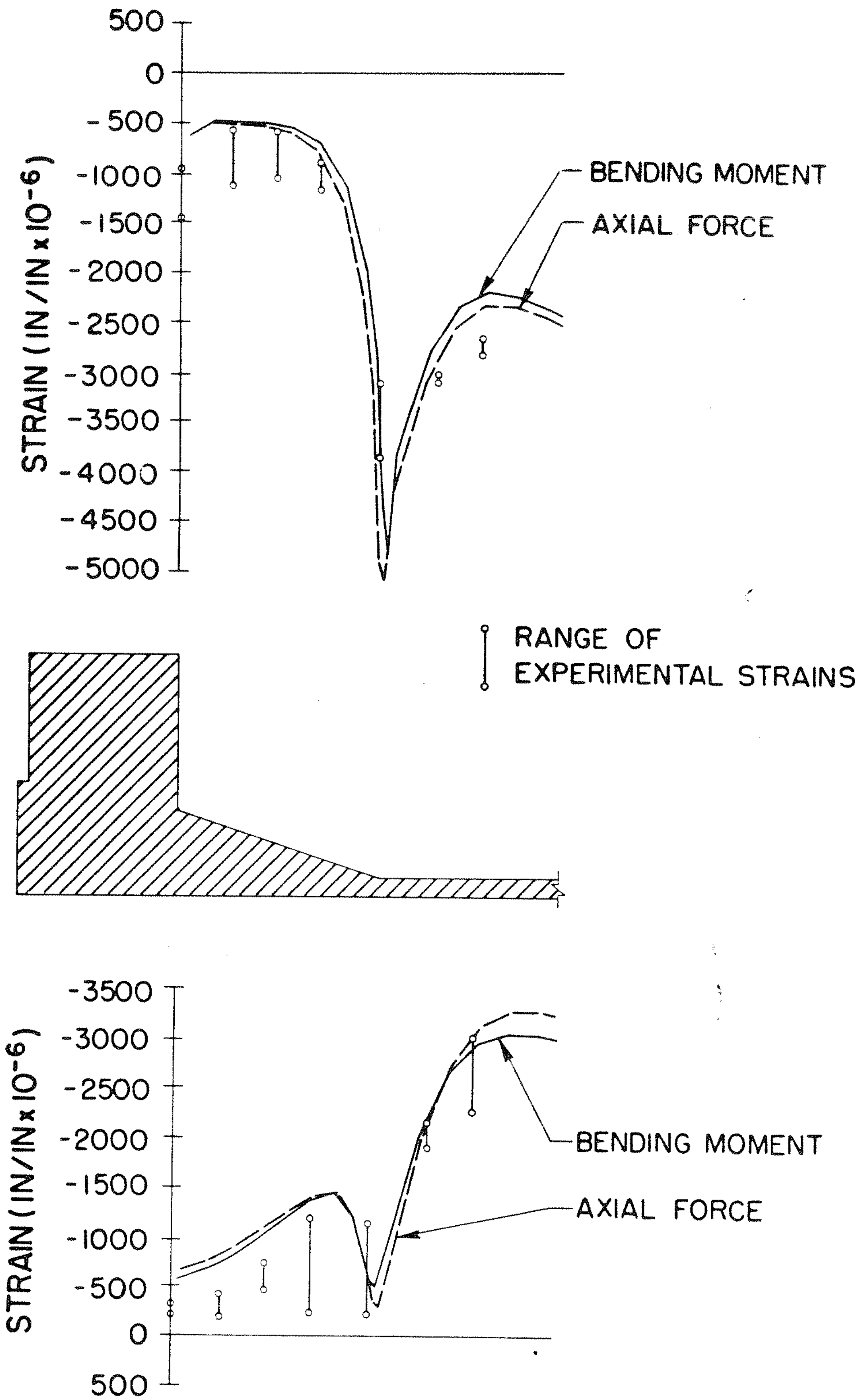

FIG. 14 SAP ANALYSIS. COMPARISON OF BENDING MOMENT AND AXIAL FORCE EFFECTS. LONGITUDINAL STRAIN.

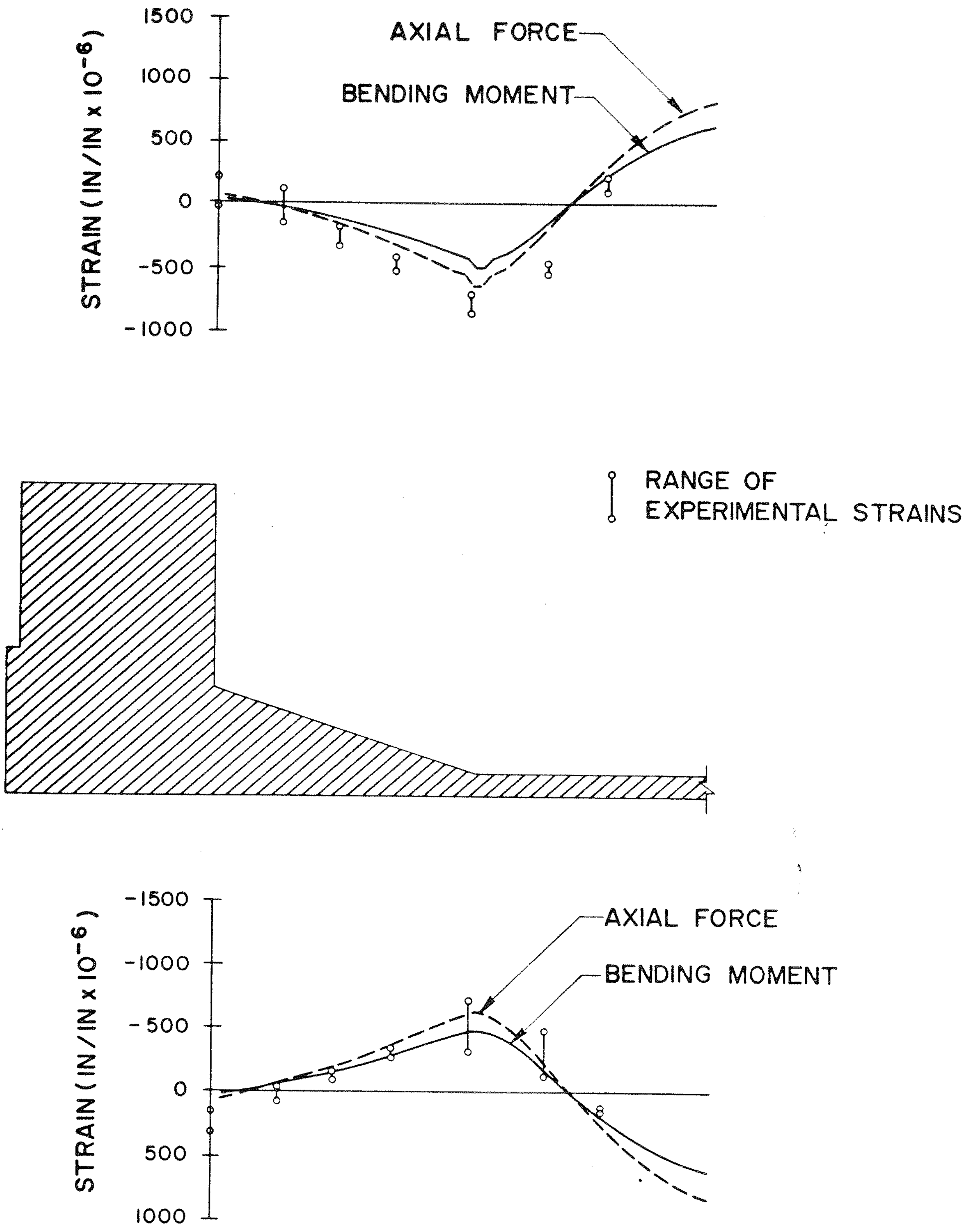

FIG. 15 SAP ANALYSIS. COMPARISON OF BENDING MOMENT AND AXIAL FORCE EFFECTS. HOOP STRAIN.

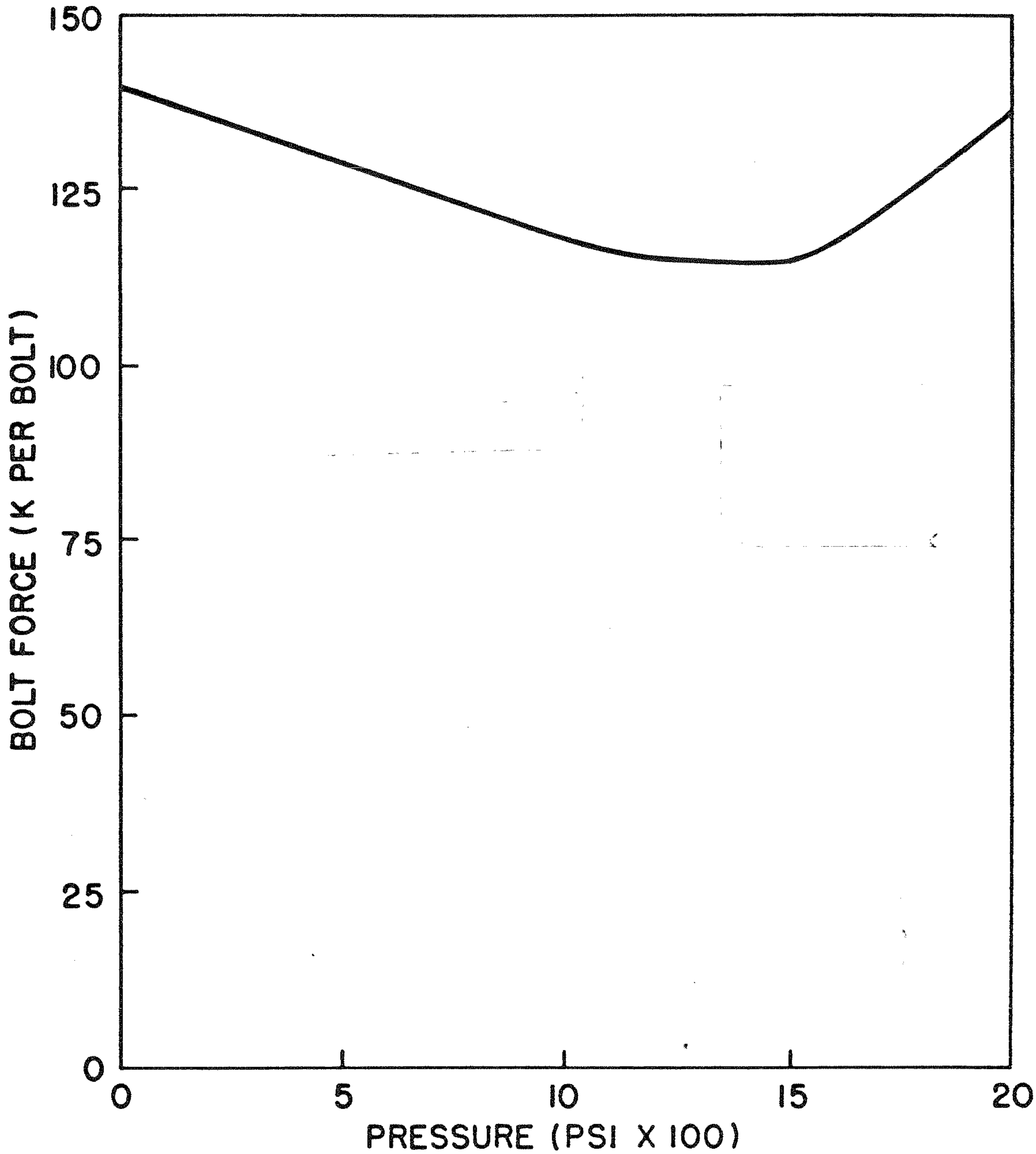

FIG. 16 VARIATION OF BOLT LOAD WITH INTERNAL PRESSURE, UNRESTRAINED PIPE

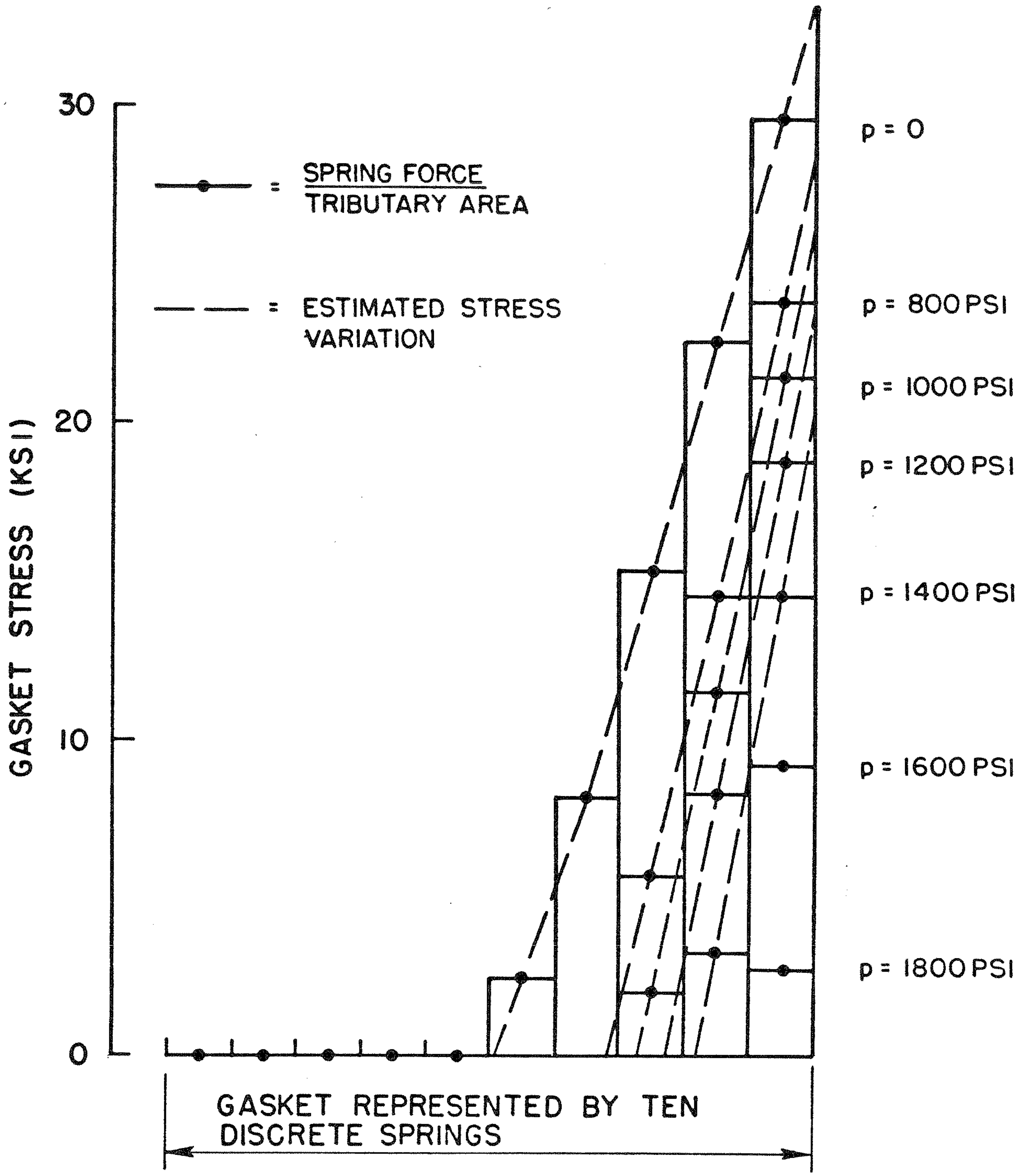

FIG. 17 VARIATION OF GASKET STRESS WITH INTERNAL PRESSURE, UNRESTRAINED PIPE

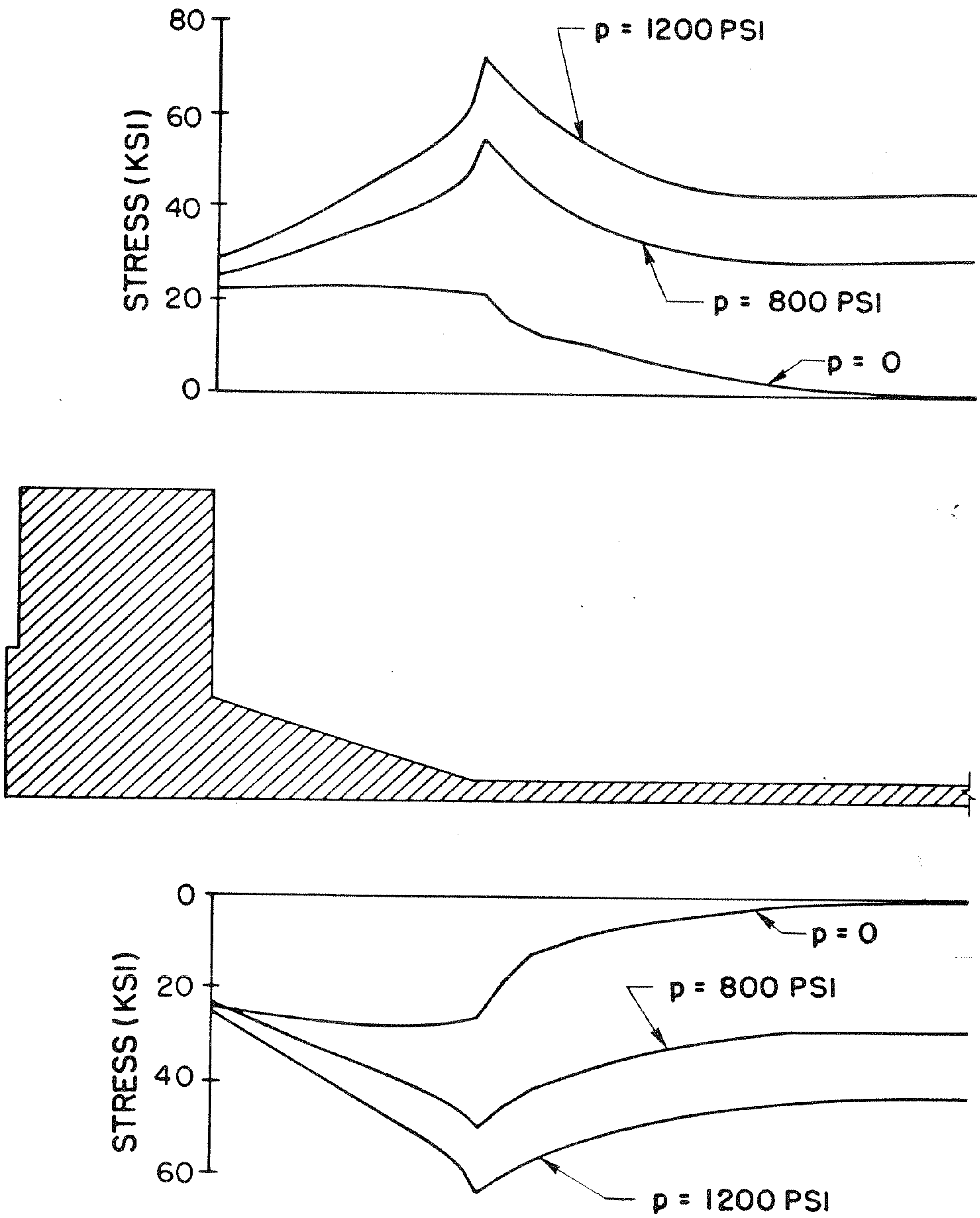

FIG. 18 FLANGE IN UNRESTRAINED LINE. VON MISES STRESSES.

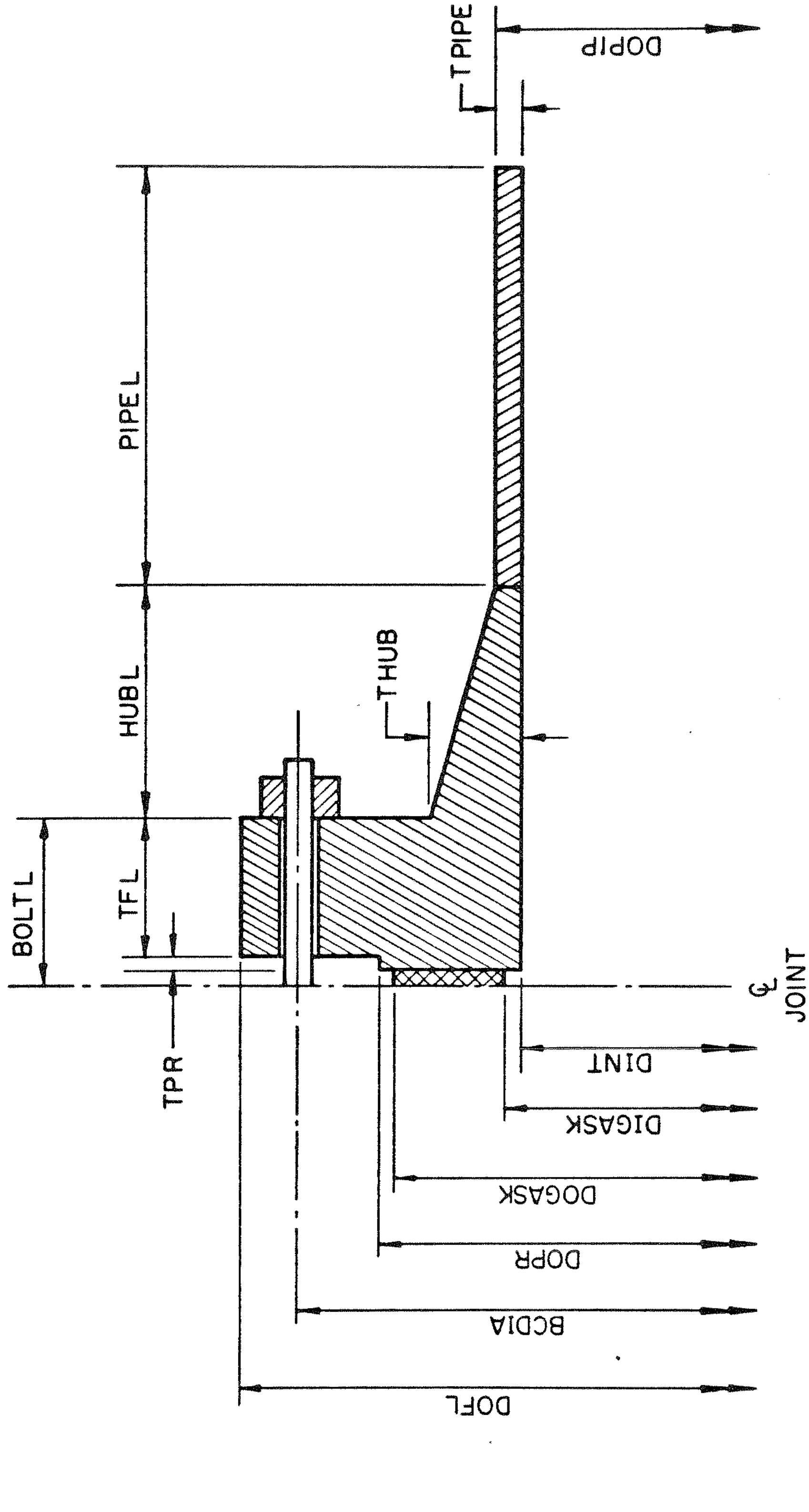

FIG. A·1 FLANGE GEOMETRY

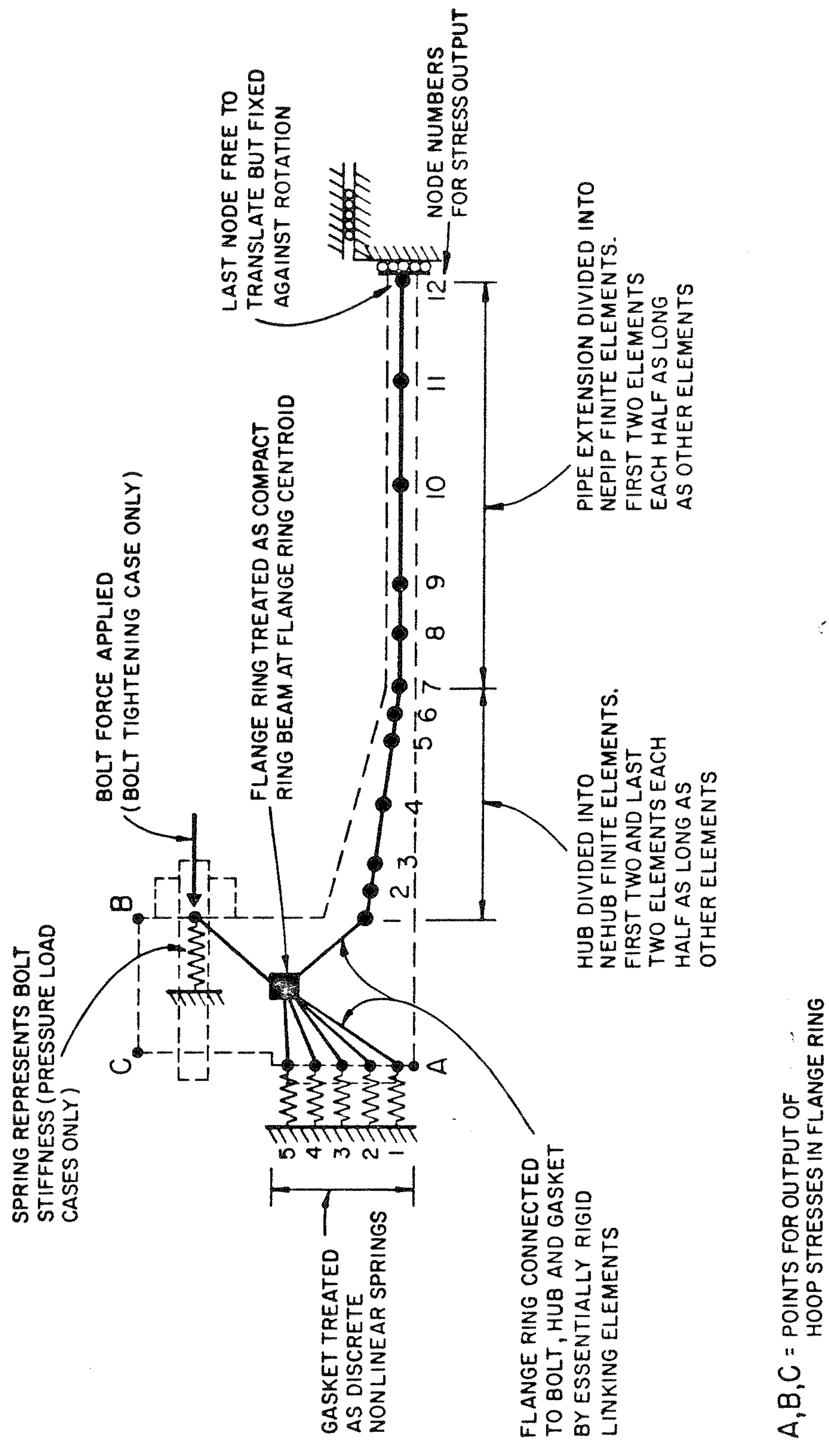

FIG. A·2 FINITE ELEMENT IDEALIZATION

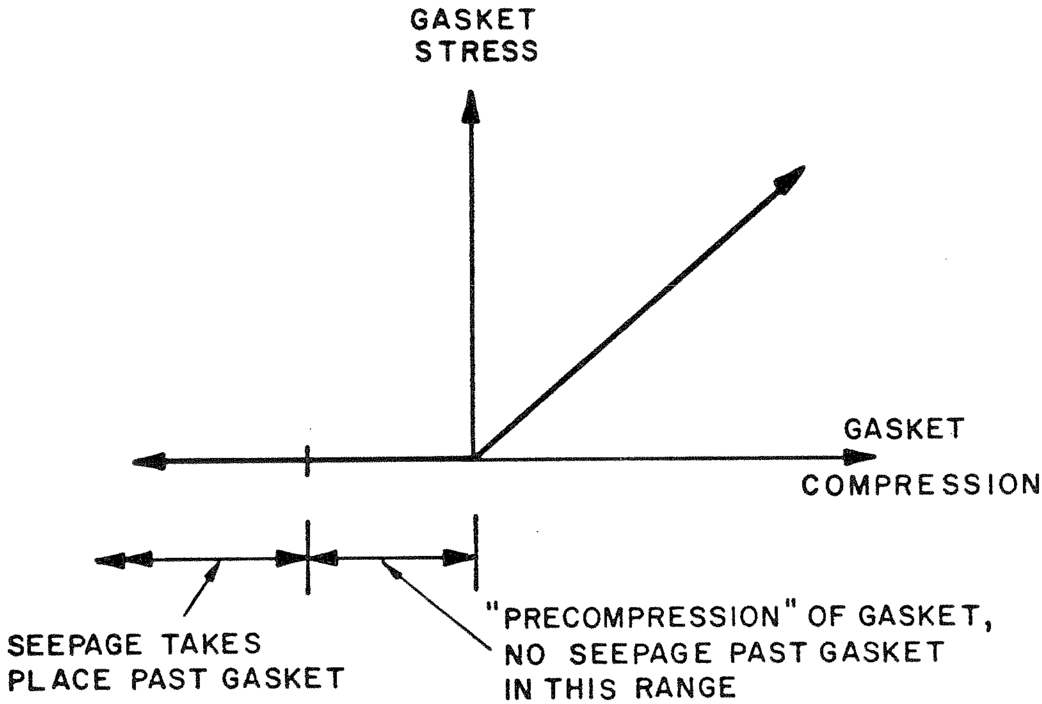

FIG. A3 STRESS-DEFORMATION RELATIONSHIP ASSUMED FOR GASKET

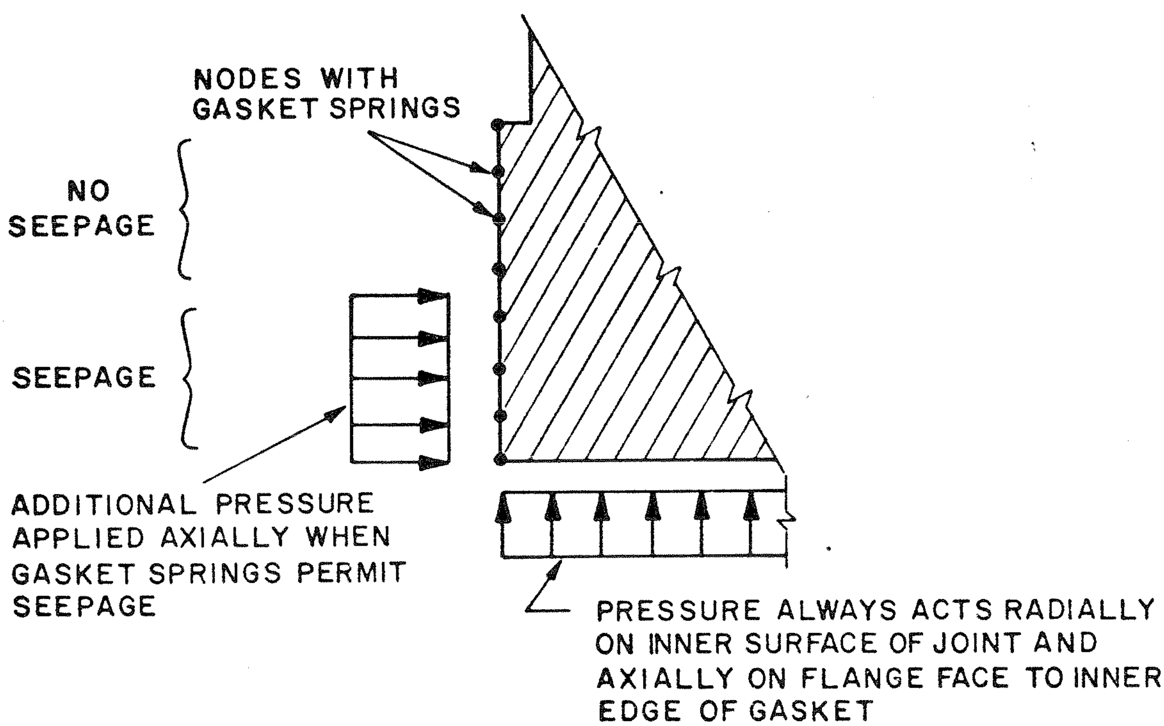

FIG. A4 ADDITIONAL PRESSURE LOAD ON FLANGE FACE

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