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Stress Indices and Stress Intensification Factors of Pressure Vessel and Piping Components

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FOREWORD

Stress intensification factors (i-factors) were first introduced into the ASA (now ANSI/ASME) Code for Pressure Piping in 1955. These i-factors were based almost entirely on cyclic-moment, fatigue tests by Markl and George[1] and by Markl[2]. The i-factors indicate the fatigue strength of a piping component (e.g., an elbow) relative to the fatigue strength of a typical girth butt weld in straight pipe. The i-factors are used to calculate stresses by equations of the form:

$$S_E = i\ (M/Z) \qquad (1)$$

where M = applied moment range and Z = section modulus of pipe. The ANSI Codes establish maximum allowable values for S_E; e.g., S_E must not exceed $f(1.25\ S_c + 0.25\ S_h)$, where f = a factor which depends upon the number of times the moments will be applied in service, S_c = cold allowable stress, S_h = hot allowable stress for the particular material and temperature.

Stress intensification factors are used in the present ANSI B31 Piping Codes; e.g., ANSI B31.1, "Power Piping" and ANSI B31.3, "Chemical Plant and Petroleum Refinery Piping". They are also used in the ASME Boiler Code, Section III, "Nuclear Power Plant Components" for Class 2 and 3 piping. Accordingly, i-factors are much more widely used than the stress indices" for Class 1 piping of ASME Code Section III.

Stress indices were introduced in the ASME Code Section III in its first edition (1963) for nozzles in pressure vessels subjected to internal pressure loading. These were derived from tests in which stresses were determined either by the use of photoelastic models or by the use of strain gages on steel models. Much of the test data available in 1963 is summarized by Mershon[3,4]. The stress is obtained from these indices by the equation:

$$S = I\ \frac{PD}{2T} \qquad (2)$$

where I is the stress index, P = internal pressure, D = vessel diameter and T = vessel wall thickness.

The concept of stress indices was broadened for Class 1 nuclear power plant piping by the addition of B, C and K indices where each was related to a different stress characteristic; specifically:

B: resistance to gross plastic deformation (limit load concepts)
C: primary - plus - secondary stresses
K: highly localized stresses (e.g., at toe of fillet weld).

The stress indices were identified with a particular type of load by subscripts: 1, for pressure; 2, for moments and 3, for thermal gradients and then incorporated in equations of the form:

$$B_1 \frac{PD_o}{2t} + B_2 \frac{M}{Z} < S_L \qquad (3)$$

$$C_1 \frac{PD_o}{2t} + C_2 \frac{M}{Z} + C_3\ E_{ab} \left| \alpha_a T_a - \alpha_b T_b \right| < S_L \qquad (4)$$

$$K_1 C_1 \frac{PD_o}{2t} + K_2 C_2 \frac{M}{Z} + K_3 C_3 E_{ab} \left| \alpha_a T_a - \alpha_b T_b \right| < S_L \qquad (5)$$

In the above equations, S_L is a stress limit assigned by the ASME Code Section III; e.g., for Equation (3), Design Conditions, $S_L = 1.5S_m$ where S_m = allowable stress intensity for the particular material and temperature. The symbols used in equations (3), (4) and (5) are defined in subarticle NB-3600 of ASME Code Section III and it is suggested that the reader consult the Code for exact definitions. Abbreviated definitions for the convenience of the reader are: P = internal pressure, D_o = pipe outside diameter, t = pipe wall thickness, M = moment or moment range, Z = section modulus of pipe, E_{ab} = average modulus of elasticity of the two sides of a gross structural discontinuity, $\alpha_a(\alpha_b)$ = coefficient of thermal expansion on side a(b) of a gross structural or material discontinuity and $T_a(T_b)$ is the range of average temperature on side a(b) of a gross structural or material discontinuity.

A correlation between i-factors and C_2K_2 is given by:

$$C_2K_2 = 2i \tag{6}$$

Since i-factors are referred to a typical girth butt weld in straight pipe, multiplying by a factor of two is equivalent to changing the baseline or reference standard to plain, straight pipe under moment loading. Accordingly, if C_2 and K_2 indices or their products are available for some piping component, such as a lateral, the i-factor can be calculated by Equation (6). The inverse process of determining C_2 and K_2, given the i-factor, is more complex since a judgment must be made as to how to apportion 2i between C_2 (primary-plus-secondary stress) and K_2 (highly localized stress). As bounding examples of this apportionment, for a girth butt weld, $C_2 = 1.0$ and $K_2 \simeq 2$; whereas, for an elbow, $C_2 \simeq 2i$ and $K_2 = 1.0$.

The Markl/George cyclic moment fatigue tests were extensive and i-factors were developed for elbows, miter bends, ANSI B16.9 tees, fabricated branch connections, concentric reducers and girth fillet welds. This test data constitutes the foundation for both i-factors and C_2K_2 indices. However, it was recognized by Markl and George that their tests covered only the most commonly used piping components isolated from other components in a piping system. The papers contained herein may be viewed as investigations of piping components, and/or interaction between piping components, which contribute to our understanding of this subtly complex subject.

The first three papers address the problem of laterals; at present, neither i-factors nor stress indices are available for such branch connections. The paper by Palusamy gives an overview of the PVRC program on laterals and a summary of completed work. The paper by Raju gives details on PVRC lateral #2. The paper by Hsiao and Khan gives data on laterals with in-plane and out-of-plane moments applied to the branch with both ends of the run fixed. The first two papers give data on internal pressure loading. This is relevant to C_1 and K_1 indices. All three papers give primary-plus-secondary stress data on moment loadings. This is relevant to C_2 indices but ultimately must be supplemented by data (or engineering judgment) regarding appropriate K_2 indices in order for i-factors to be obtained by Equation (6).

The fourth paper, by Bryson, Johnson and Bass, addresses the complex problem of nozzles in vessels or piping; complex not only because of geometry but also because of the seven independent loadings. The results are both significant and timely in that:

(a) For pressure loading, the present C_1K_1 product of $1.5 \times 2.2 = 3.3$ is shown to be conservative for nozzles with Code-required reinforcing.

(b) For moment loading on the branch, the present $K_{2b}C_{2b}$ product is appropriate.

(c) For moment loading on the run, the present $K_{2r}C_{2r}$ product is too conservative.

The fifth paper, by Karabin, Mello and Hulbert, discusses the complicated geometry in which a branch connection (Sweepolet ^{R}) is placed in a short length of pipe between two elbows. An in-plane moment applied to the elbows produces significant stresses in the vicinity of the branch connection. The subject of flexibility factors of interacting components is also addressed in detail.

The sixth paper, by Avent and Sadd, gives data on eccentric reducers; i-factors and stress indices are given in Codes for concentric reducers but not eccentric reducers. The results are relevant to C_1 and C_2 indices. The authors chose to define B_1 and B_2 indices as the maximum membrane stress divided by the nominal stress PD/2t for pressure and M/Z for moment, however, this is not equivalent to limit load concepts used in developing other B_1 and B_2 indices. The problem is that membrane stresses, alone, do not govern the occurrence of gross plastic deformation, for example, a cantilever beam with an end load has zero membrane stress, but a limit load is readily calculable. The authors chose to define C_3 as the maximum membrane plus bending stress divided by $E\,\alpha|T_2 - T_1|$ where E is the modulus of elasticity, α is the coefficient of thermal expansion, and $|T_2 - T_1|$ is the instantaneous temperature change on the inside surface. However, as indicated in Equations (4) and (5), C_3 as used in ASME Code Section III is associated with a gross structural or material discontinuity. The authors' results are perhaps more relevant to the ΔT_1 and/or ΔT_2 terms used in the Code equation for S_p. In any case, the reader should be certain that he understands the basis of the B_1, B_2 and C_3 indices before using them in Code equations.

The papers by K. Thomas and A.K. Dhalla address "end effects" on elbows. The theory on which i-factors and C_2 stress indices in the Codes are based assumes that nothing is attached to the ends of the elbow. Actually, the stresses and flexibility factor of elbows may be greatly affected by what is attached to the ends of the elbow. The Markl/George papers include results of tests on elbows of various arc angles, showing that as the arc angle decreases the i-factors decreased. The ANSI B31 Codes and the ASME Code Section III for Class 2 and 3 piping give correction factors to the i-factors for use when a flange is attached to one or both ends of an elbow. The papers by Thomas and Dhalla provide valuable additional guidance on elbow end effects. These papers also consider the stress or strain fields at welds between elbows and straight pipe.

The paper by Sidorowicz addresses the question: What are the stresses when a lug support is used on a straight pipe? There are no i-factors or stress indices for lugs; although ASME Code Case 122 gives (highly conservative) guidance for such integral structural attachments. The authors chose to use an allowable stress of 46.7 MPa (6770 psi). This is an allowable membrane stress for the material and design temperature. The question arises: is this an appropriate stress limit for the complex type of stresses that exist in the vicinity of a lug? Indeed, calculation of stresses is only part of the stress-based design process. An appropriate limit to the calculated stresses must be established and, in doing so, the potential failure mechanisms must be considered.

The final paper by Moore and Rodabaugh discusses the background of B_1 and B_2 "stress indices" which could more accurately be called "limit load indices".

The collection of papers herein give a good indication of the state of the art of stress indices and stress intensification factors. They also implicitly indicate both the motivation for establishing stress indices and i-factors and a pitfall in their use. The motivation is to reduce the design evaluation cost. The pitfall is that, in using simple yet broad-in-scope design evaluation methods, construction costs may increase. An appropriate balance is desirable but not easy to achieve.

Readers should recognize that the papers do not contain the equivalent of Code-approved evaluation methods and should apply careful engineering judgement in making use of any information given in the papers.

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In all fairness to the readers, we wish to point out the obvious, that is, the papers in this Journal represent the opinions of the authors. Likewise, in fairness to the authors, it is noted that the Journal was printed in advance of the Session and the opportunity to adjust or clarify a point of view based on questions and discussion did not exist.

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CONTENTS

PVRC LONG RANGE PLAN FOR THE DEVELOPMENT OF STRESS INDICES FOR 45° BRANCH CONNECTIONS

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ABSTRACT

Early in 1974, the PVRC subcommittee on Reinforced Openings and External Loadings set up a Task Group on Laterals. The objective of this Task Group is to carry out state-of-the-art studies to understand the behavior of lateral pipe and nozzle connections. A more specific and short range objective is to develop stress indices for laterals. Since its inception the Task Group has accomplished the following: A survey of existing geometric configurations and loadings have been carried out. A long range plan has been defined. After examining alternative methods of analysis, the three-dimensional finite element technique has been chosen and a validation of this method has been carried out based on the photoelastic model test results. Finite element analysis of two models have been completed for internal pressure as well as in-plane moments on branch and run pipes. Analysis of a third model is in progress. This paper presents the details of the development plans and discusses the progress made to-date.

INTRODUCTION

The fatigue evaluation procedures specified in Section III (for Class 1 components, and Section VIII Division 2 of the ASME Boiler Code* [1] requires the knowledge of peak stresses in a given component. Until recently, no satisfactory fully analytical method was available for the determination of peak stress in complex geometries such as lateral pipe or vessel connections [2]. Although, with the advent of three-dimensional finite element analysis, a fully analytical method is now available, it is not always feasible or cost-effective to make a finite element analysis for each application. In the past when no fully satisfactory analytical method was available, the code recommended a simple and a conservative method called the stress index method which is still useful for the reasons cited above. The stress index method simply consists in expressing the peak stress as a product of a multiplier called stress index and a known reference stress component such as the nominal hoop stress of a main pipe. The present code rules provide values of stress indices for various openings in vessels [2] and connections in pipes [3]. Except for pressure loading normal stress index estimate specified in [2] for branch pipe (nozzle)[1] to main pipe (vessel) internal diameter ratio less than 0.15, no other stress index values are provided in the code for the lateral connection.

Realizing the need for the lateral stress indices, early in 1974, the subcommittee on Reinforced Openings and External Loadings (ROEL) of the Pressure Vessel Research Committee (PVRC) set up a Task Group on Laterals. The long range objective of this Task Group is to carry out state-of-the-art studies and carefully planned experiments with the help and participation of pressure vessel and piping industry and recommend a complete set of stress indices for inclusion in the appropriate sections of the code. As a first step, the Laterals Task Group has been performing exploratory studies with the objective of defining the scope of the problem and to call attention to areas of concern towards verifying the existing designs and proposing new rules.

The purpose of this paper is to describe the present phase of exploratory studies, summarize progress made to-date and discuss plans for future work. A very brief discussion of the stress index method is presented. The overall plan of the present phase of work is described. Some key results obtained for one unreinforced photoelastic model and two reinforced steel models are presented. Future scope of work and tentative plans for cooperation between industry and the Pressure Vessel Research Committee (PVRC) are discussed.

STRESS INDEX METHOD

The stress index method is based on the simple idea of expressing the peak stress components by the product of a multiplier called stress index and some chosen nominal reference stress component that can be calculated readily for a given component. The multiplier, stress index, is then defined as the ratio of the peak stress component value to a chosen readily calculable membrane stress component. While the basic idea remains the same there may be minor differences in the specific definition of stress index depending on the choice of reference stress in various parts of the code. In a given application the designer/analyst must refer to the applicable paragraph of the code.

For the purposes of discussion, let I be the stress index and σ be the reference stress. Following the general practice followed in the code for the reinforced openings and connections [2], let σ be the membrane hoop stress for the corresponding unpenetrated vessel or main pipe material.

$$\sigma = \frac{P(D + T)}{2T} \tag{1}$$

where D and T are the inside diameter and thickness, respectively and P is the internal pressure. The components of stress that are of interest for a nozzle/pipe connection are, as shown in Fig. 1, the tangential (σ_t), normal (σ_n) and

* Hereafter code will refer to ASME Boiler Code

[1] For brevity, reference to vessel nozzle connections will be omitted in the sections to follow.

radial (σ_r) components. The fourth quantity that is of interest is the stress intensity S, defined by

$$S = \text{Maximum of } [|\sigma_1 - \sigma_2|, |\sigma_2 - \sigma_3|, |\sigma_3 - \sigma_1|] \tag{2}$$

where σ_1, σ_2 and σ_3 are the principal stress components.

Given the definitions of equations (1) and (2), let the following definitions for the various stress indices be chosen:

$$I_n = \frac{\sigma_n}{\sigma} \tag{3}$$

$$I_t = \frac{\sigma_t}{\sigma} \tag{4}$$

$$I_r = \frac{\sigma_r}{\sigma} \tag{5}$$

$$I = \frac{S}{\sigma}$$

For a given geometry (D and T) and loading (P), the refernce stress can be calculated by equation (1). If the values of stress indices, I_n, I_t, I_r and I are known, then it is a simple exercise to compute the corresponding peak stress components ($\sigma_n, \sigma_t, \sigma_r$) and stress intensity (S). Therefore, the ability to perform a stress index method calculation simply depends on the availability of stress index values for the geometries and loadings of interest.

In the case of pipe connections normal to the run pipe the code provides stress indices for a wide range of geometries and loadings [2,3]. However, in the case of lateral connections the code does not provide stress indices that can be of practical use. Only stress index contained in the code on lateral refers to an estimate of the inside value of I_n under pressure loadings for d/D less than or equal to 0.15. The estimate is given by the relation:

$$I_n = I_n^* \; [1 + (\tan \phi)^{4/3}], \tag{7}$$

where

I_n^* = the inside value of I_n for a radial connection as defined in [2]

ϕ = included angle between the axes of branch pipe and run pipe, Fig. 1.

The above estimate is inadequate because it does not cover many cases of practical interest in piping (e.g., d/D = 0.50).

In the case of inplane moment loadings on the run (M_{IR}) and branch (M_{IB}) pipes, the reference stress is defined as:

$$\sigma = M_{IR}/Z_R \tag{8}$$

$$\sigma = M_{IB}/Z_B$$

where Z_R and Z_B are the section moduli of run and branch pipes, respectively.

EXPLORATORY ANALYSIS PLAN

At the outset early in 1974 it was realized that the scope is large and the available PVRC funding is limited. Therefore, careful choices had to be made. An informal survey of the configurations manufactured and used by the industries and those found in the literature showed that 45 degree lateral conditions are by far the mostly used geometry. The primary geometric parameters are the ratio of branch pipe inside diameter to run pipe inside diameter (d/D), the ratio of run pipe inside diameter to its wall thickness (D/T) and the ratio of branch pipe nominal hoop stress to run pipe nominal hoop stress (s/S). The above referenced survey indicates that a nominal hoop stress ratio value of unity (s/S = 1) is of most practical interest whereas in the case of d/D and D/T ratios, the ranges of most practical interest are 0.08 to 0.5 and 10 to 40, respectively. The code provides several choices of reinforcement configurations (for example, see [3]) and the generally referenced standard reinforcement configuration (See Fig. NB-3643-3(a)-2 [3]) was chosen as the configuration of most practical interest.

With the above mentioned choice of parameters four models were chosen for the exploratory study. Fig. 2 shows the longitudinal cross-section of the models. The various geometric parameters are defined in this figure. Table 1 contains the dimensionless ratios of geometric parameters of models 1 through 4. The loadings to be considered are internal pressure (P), inplane moment on the run pipe (M_{IR}) and inplane moment on the branch pipe (M_{IB}).

METHOD OF STRESS INDEX DETERMINATION

On evaluating various alternative methods of analysis, both analytical and experimental, that can be used in the determination of stress index for laterals, the Laterals Task Group decided that the most economical way to proceed is to use three-dimensional finite element analysis technique. Among the factors that favor finite element analysis are the capability to obtain directly information on displacements, the ability to visualize the behavior through distorted geometry plots and the ease with which a number of Loads can be analyzed on a given model [4].

QUALIFICATION OF FINITE ELEMENT TECHNIQUE

Due to differences in modeling assumption and availability of several different finite elements, it was felt necessary, as a first step, to carry out a qualification study to assess the capability, accuracy and cost of three-dimensional finite element technique. For this purpose, photoelastic model WC-12B2, previously tested by Leven [5] was chosen. Two investigators (Swanson Analysis Systems, Inc., Elizabeth, Pa. and Hay and Associates, Limited, London, U.K.) were chosen to perform two independent three dimensional analyses of the photoelastic model. The detailed results of their study are contained in the reports submitted to the PVRC [6,7] and a summary and comparison of finite element and photoelastic results are available in the Welding Research Council (WRC) Bulletin 251 [4]. A brief summary of this qualification study is presented in the following.

The geometry of model WC-12B2 is shown in Fig. 3. The vessel is 29.68 inches long and its internal diameter and thickness are 6.374 inches and 0.523 in., respectively. The nozzle, intersecting the vessel longitudinal axis at 45 degrees, is 6.864 in. long and its internal diameter and thickness are 0.820 in. and 0.448 in., respectively. The juncture external fillet radius is 0.175 in. The Elastic Modulus and the Poisson's ratio for the model material are 6510 psi and 0.485, respectively. It is subjected to an internal pressure loading of arbitrary value. The values of dimensionless parameters d/D, D/T and s/S are 0.13, 12.2 and 0.15, respectively. If s/S is assumed equal to unity and if the available area for reinforcement is computed, then it turns out that the lateral can be assumed to be fully reinforced. As pointed out before Swanson Analysis Systems, Inc. (SASI) and Hay and Associates, Limited (HAL) performed two independent analyses. Some of the key details of the analysis and results are compared in Table 2. SASI modeled the photoelastic lateral using 8 node isoparametric incompatible displacement elements with which the number of integrating points can be varied from 8 to 27. The elements in the juncture region were chosen to have 27 integration points whereas the regions removed from the juncture were chosen to have 8 points of integration. Taking advantage of the symmetry and stress attenuation characteristics, SASI modeled about one quarter of the lateral. An overall view of the SASI finite element model is shown in Fig. 4. HAL used 20-node isoparametric brick elements. Since HAL could not get a convergent solution for the Poisson's ratio value of 0.485, HAL produced a solution for a Poisson's ratio value of 0.3. The peak stress index was then corrected for the Poisson's ratio difference using a two dimensional parametric study. On the other hand SASI obtained a complete stress and displacement solution for both Poisson's ratio values of 0.485 and 0.3. Typical stress distributions obtained by SASI are shown in Figures 5 and 6 in which the results of photoelastic study [5] are plotted. Similar stress distributions are contained in [4] for various locations in the longitudinal and lateral cross-sections of the photoelastic lateral. Based on the comparison of finite element and photoelastic results [4] the PVRC Laterals Task Group reached the following conclusions with respect to the applicability and usefulness of finite element analysis:

- o Given the current state-of-the-art, there is no simple, inexpensive way to judge the accuracy of finite element surface stress values.
- o Good finite element results can be obtained by a competent, investigator if he has adequate bench mark data for comparison. Alternatively, confidence in results can be established by other means such as parametric and mesh convergence studies, but these studies may increase the cost of the analysis significantly.
- o If one does not have the means to carry out the supportive studies, it is observed that he should carefully examine and take into account the influence of factors such as details of modeling at regions of stress concentration, the convergence with respect to element size and methods of extrapolation of stresses to the surface on the accuracy of finite element result.

On the basis of these findings, the Laterals Task Group made the following recommendations:

- o Continue lateral studies using the finite element analysis method, with particular care given for modeling and thorough examination of results.
- o Carry out parametric studies using simpler models.
- o Carry out mesh convergence studies.
- o Investigate the need and feasibility of carrying out additional photoelastic or steel model tests to establish the applicability of calculated peak stress indices for practical design.

Based on the above conclusions, the Task Group decided to proceed with finite element model studies.

FINITE ELEMENT MODEL STUDIES

The finite element analysis of the expoloratory models identified in Table 1 started in October 1976. As of January 1981, analysis of models 1 and 2 were completed and the analysis of model 4 is in progress and expected to be completed in summer of 81. Table 3 shows the present state of progress together with the type of loading considered for each model.

Each of models 1 and 2 were analyzed separately under internal pressure (P) and external inplane moment on the run (M_{IR}) and on the branch (M_{IB}) pipes. Model 1 was entirely analyzed by Raju and the results are contained in [10-12]. Model 2 under internal pressure (P) and inplane moment on branch pipe (M_{IB}) was analyzed by SASI and the results are contained in [13]. Raju analyzed model 2 under internal pressure (P) and external inplane moment (M_{IR}) on the run pipe and the results are contained in [14,15]. Model 4 is being entirely analyzed by Raju. Model 2 pressure loading results obtained in [13] were found to be inconsistent with the estimates based on experimental results and therefore this case was reanalyzed in [14].

A typical finite element model developed for model 1 by Raju [10] is shown in Fig. 7 which contains an isometric view of the entire model and a magnified view of the element layout in the highly stressed acute corner region. Details of all the models developed thus far are shown in Table 4. On the basis of comparison of results obtained by various models, it has been found that the models developed by Raju are adequate for the purposes of obtaining stress index.

Resutls obtained from each loading includes magnified plots of deformed shape, contour plots of maximum and minimum principal stresses and maximum shear stress components for various regions, stress distribution plots for selected cross-sections and tabulations of important stress components and stress indices. The computer printout containing the listing of centroidal and nodal stress components for each of the elements and nodal displacements are retained for future analysis. A typical plot of the maximum principal stress contours in the acute corner region of model 1 under internal pressure is shown in Fig. 8. The values indicated for the contours are ratios of maximum principal stress with respect to the nominal hoop stress defined in equation (1).

Based on the results obtained thus far, three regions in the longitudinal plane of the lateral can be identified to be critical and they are indicated in Fig. 9 to be acute corner region, obtuse corner region and lateral transition region. The maximum stress index due to pressure invariably occurs at the acute corner region whereas the maximum stress index due to inplane bending moment on branch occurs at the lateral transition region. For Model 1 the stress index due to inplane moment on the run pipe reached the largest value at the obtuse corner region for the longitudinal plane. However, the maximum for this loading occurred in a plane 45 degrees removed from the acute corner side of the longitudinal plane. The location and magnitude of maximum stress index (I) values associated with the stress intensity in the longitudinal plane are shown in Fig. 10. The stress index I is given by equation (6) where σ is defined by equations (1), (8) and (9), respectively for the internal pressure and inplane run and branch moments, respectively.

LONG RANGE PLAN

The plan for the immediate future is to complete analysis of models 3 and 4 and analyze and summarize all the results in a WRC bulletin. The primary objective of the summary is to identify the critical locations with respect to each of the loadings and to obtain values of stress indices (I, I_n, I_t, I_r) for each location due to each of the independent loadings in a tabular form such as, for example, shown in Table 5. This table contains I stress index values for model 1 and similar table will be developed for other stress indices (I_n, I_t, and I). From such a table, one can obtain the necessary code vessel type (2) and pipe type (3) stress indices for the given geometry and loadings. In this bulletin, the Task Group will consider recommending an interim stress index to ASME Boiler Code.

Based on a critical evaluation of results obtained for models 1 through 4 a full scope study consisting of finite element analysis and carefully planned experimental model test will be proposed. Experimental and analytical results available in the open literature and those provided by the industry will be considered in proposing a full scope study. The Task Group plans to approach the industry for participation and cooperation because at the current rate of PVRC funding it will take more than a decade to obtain a comprehensive set of indices for recommendation to the ASME Code.

SUMMARY AND CONCLUSIONS

The PVRC Task Group on Laterals was set up by the subcommittee on Reinforced Openings and External Loadings in early 1974 and its long range objective is to recommend a set of stress indices to the ASME Code. After conducting a survey and evaluating alternative methods, the Task Group has defined four lateral models for exploratory study and has chosen three-dimensional finite element analysis technique for the determination of stress indices. A finite element analysis of a photoelastic model performed to qualify the finite element method showed that good results can be obtained by careful modeling. The finite element analysis of two models have been completed and the remaining two models are expected to be completed in 1982. At that time a summary report will be prepared for WRC publication in which the results of the exploratory model study will be critically evaluated and a full parametric study will be proposed. At that time the Task Group will consider recommending an interim stress indices to the code. The cooperation and participation of industry will be sought for a speedy determination of a comprehensive set of stress indices for recommendation to ASME Code.

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Model	D/T	d/D	L_n/T	T_n/T	t/T	r_3/T	T	r_1/T	r_2/T
1	10	0.08	2.0	0.782	0.08	0.50	3	0.5	1.0
2	10	0.50	2.0	1.733	0.50	0.87	3	0.5	1.0
3	40	0.08	2.0	1.033	0.04	0.70	0.75	0.5	1.0
4	40	0.50	4.0	3.466	0.50	2.10	0.75	0.5	1.0

NOTE: (1) r_1 & r_2 are to be constant around Branch Axis in Planes normal to the Branch Axis

(2) Each model are to be analyzed under independent loadings of internal pressure (P) and external inplane moments on run (M_{IR}) and branch (M_{IB}) pipes.

TABLE 1 LATERAL MODEL GEOMETRY CHOSEN FOR THE PVRC LATERAL TASK GROUP STUDY

Analysis Detail	Swanson Analysis Systems, Inc.	Hay & Associates, Limited	Photoelastic Results
Element Type in the Remote Region	8 Node Isoparametric Incompatible Displacement	20 Node Isoparametric	
Computer Program	ANSYS [8]	PAFEC70 [9]	
Number of Elements	1020	154	
Poisson's Elements	0.485, 0.3	0.3	
Peak hoop stress at the acute corner for Poissons' Ratio = 0.3	4.05	2.9	4.5
Peak hoop stress at the acute corner for Poisson's Ratio = 0.485	3.6	3.5	

TABLE 2 COMPARISON OF PHOTOELASTIC AND FINITE ELEMENT ANALYSIS