A PARAMETRIC GROSS DISCONTINUITY INTERACTION FINITE ELEMENTS ANALYSIS BETWEEN ADJACENT NOZZLES OF A VESSEL

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SUMMARY

The aim of this paper is a brief discussion of the effect of nozzles interaction in a shell in terms of the gross discontinuity stresses. However, while the peak stress distribution in an isolated nozzle has been largely studied in the open literature, there are not many contributions in this particular field. ASME III code serves as a general reference and its nozzles interaction rules will be compared with the results obtained by this analysis. Besides it is the principal aim of this paper to give a critical contribution in order to study the cases where simplified ASME Codes procedures are not applicable, without drawing a quite general conclusion on this topic. The paper is divided into three sections: analytical tools, a parametric analysis of a few cases, a comparison with ASME III code provisions.

a) *Analytical tools:* to perform this analysis a chain of programs were coded; they include: — CIGEME: an automatic mesh generator for the case of two intersecting cylinders; two types of mesh may be generated depending on radius ratio. The mesh may have variable size, being rougher for point far apart from the nozzle. — RAMESH-RANUN: these two connected programs combine the meshes as generated by CIGEME, in order to have a single mesh for the two intersecting nozzles. The minimization of the stiffness matrix bandwidth is the primary aim of these two programs. — LAUDA: all the load data and the remaining geometrical ones are fed to this program in a compact form and exploded in a format compatible with the stress analysis program. — SAP 4: this is a general finite element program; the shell element has been selected to perform this analysis; the boundary conditions partially coming from the structural symmetries are introduced by means of the boundary elements. The chain of the programs has been held as flexible as possible, so that each unit can be used both as an indipendent program or part of the chain. Besides plotter display is used to check the model and to visualize the mesh distribution.

b) *Parametric analysis:* all the examined cases are relative to a thin shell. The parameters which are investigated by means of a few groups of different models are: 1) nozzle diameter to vessel diameter, 2) distance between adjacent nozzles. In the results examination the effect which was particularly investigated was the amount of perturbation due to a second nozzle on the stress-strain pattern with reference to the one due to an isolated nozzle under the same loading conditions.

c) *ASME III compliance:* a reference is made to the provisions of ASME III NB; it is found that a substantial agreement exists between the present results and ASME III provisions. Some indications are given on the stress distribution in the ligament region between the two nozzles for cases when the stress index method is not throughly applicable.

1. Introduction

The evaluation of the stress singularities around a nozzle is a typical problem of the analysis of reactor pressure vessels designed to meet the requirements of ASME 3 code Nuclear Components [5] . Many studies both by the use of the classical theory of shells and by the use of F.E. tecniques have been published (one may quote as examples ref [8] , [9] , [10] , [11] , [12] as many others), while some experimental results are available (see ref [3] as an example). However some of these studies are relative to the stress concentration around a hole, more than to the study of the stress state in a nozzle.

The same circumstance may be noted if the problem of the interaction of two neighbouring cylinders is considered; Peterson [1] has discussed the problem of two holes interaction both in a plate and in a shell. In particular Peterson [2] , quoting results by Chi-Bing-Ling[7] , has shown that in a plate for the case of uniaxial stress state as well as for the case of a biaxial state with equal components, the mutual influence of the two holes is very limited provided that the ligament length is larger than roughly two times the diameter. However Hamada et al [6] have shown that for the case of a shell the same considerations can be applied with the only difference that the S.C.F. are higher.

As far as ASME 3NB code is concerned, exclusion rules are formulated in terms of the distance of the two nozzles centerlines, as follows :

$$
(1a,c) \quad \begin{cases} \delta \geq 1.5(d_1+d_2) & \text{for longitudinal direction in cylinder} \\ \delta \geq (d_1+d_2) & \text{for circumferential direction in cylinder} \\ \delta \geq (d_1+d_2) & \text{for spheres} \end{cases}
$$

δ distance between nozzles centerlines

d_i nozzle diameter

If inequalities (1 a,c) apply the two nozzles may be considered as indipendent and this is generally the case for pressure vessels. However the authors found a case where these inequalities did not hold and no distance increase could be performed due to lay out difficulties, consequently interaction had to be studied.

Further due to the presence of nozzles a F.E. analysis tecnique was used, to compute the stresses and strains in the ligament region. Eventually a limited number of cases were examined with the aim of collecting data about this particular phoenomenon.

2. Analytical tools

While SAP4 [4] program was selected to perform the stress analysis, some effort was necessary to keep the work for input preparation and handling within reasonable limits. To do so, a chain of program was made as indicated in table 1.; it includes :

CIGEME : an automatic mesh generator for the case of two intersecting cylinders. The mesh type can be decided by the user to meet his particular problems; obviously the usual practise of having a mesh much more refined around the nozzle, than at large distances can be met by the code.

RAMESH-RANUM : these two interconnected programs can couple the meshes generated for the case of two isolated nozzles; reordering and renumbering of the nodes is performed so that the stiffness matrix bandwidth is reduced to a minimum.

LAUDA : the load input data, as well as other geometrical (thickness) data are fed in a compact form and exploded by the program as compatible for SAP4.

For all these cases a plotting facility was available to assist the analyst in the data preparation and control. In fig. 1 typical plots of the meshes for these analyses are shown; as one can see an assionometric type of representation was selected for this case.

In the preparation of the data much care was given to a correct representation of the boundary conditions for a correct representation of the symmetries the use of which was found necessary to keep the size of the problem within a reasonable size. Some problem was found in representing the correct behaviour of the nozzle quarter, which could, to a limited extent, have influenced the results. Eventually a compromise between simplicity and accuracy was obtained, assuming that (see fig. 1)

a) the correct membrane displacements were given to the upper and lower boundary (along the z axis)

b) compenetration of the two cylinder halves was not allowed on the left boundary imposing the appropriate symmetry conditions

c) radial displacement only was allowed on the right boundary

d) a symmetry condition was assumed for the model centerline as, for semplicity sake, the two nozzles were assumed to have equal sizes

e) a condition for the conservation of plane sections was assumed at the nozzles terminals.

3. Parameteric analysis

The following cases have been analyzed :

TABLE 2

CASE	NOZZLES NUMBER	CYLINDER Dia D(m)	CYLINDER Thickn. T (m)	NOZZLE Dia d (m)	NOZZLE Thickn. t (m)	NOZZLE DISTAN.	δ/d	ASME 3 EXCLUS.
R 150	2	4.0	0.1	0.30	0.09	.80	2.65	NO
R 300	2	4.0	0.1	0.60	0.09	1.60	2.65	NO
R 400a	2	4.0	0.1	0.80	0.09	2.00	2.50	NO
R 400b limit	2	4.0	0.1	0.80	0.09	2.40	3.00	YES
R 400c	2	4.0	0.1	0.80	0.09	2.80	3.50	YES
R 500	2	4.0	0.1	1.00	0.09	2.40	2.40	NO

For every case one finds the main dimensions of the models as well as the result of A-SME 3NB exclusion rule.

Further the following general considerations are applicable for each model :

a) the nozzles diameters are the same

b) both the nozzle centerlines are placed on the same cylinder meridian

In all the cases the membrane stress field produced by the pressure is assumed : $\sigma_h = 14\ Kg/mm^2$ $\sigma_z = 7\ Kg/mm^2$ σ_h and σ_z being the hoop and axial stress.

As indicated, many of the analyses do not meet the exclusion rules as suggested by ASME 3, besides if one considers the adimensional main parameters :

$$B_1 = d/D \quad B_2 = \delta/d \quad B_3 = T/D \quad B_4 = t/T$$

one finds that the parameter B_2 only is retained by ASME 3 in the analysis of the stress state indipendence between the two nozzles. In the present case the following parameters are considered as a constant :

B_3 for all the cases

B_1 for the cases R400a, b, c

while B_2 is roughly a constant for the cases R150, R300, R400a, R500

As far as B_4 parameter is concerned some variations exists from case to case, however the importance of this parameter, at least following Seika et al. [3] paper, should not be decisive.

Further one may note that again following Seika's paper, one parameter is not explicit in this analysis i. e. the ratio ρ / T (ρ being the curvature radius at the nozzle base), this is due to the very nature of the model which is used in the present case (thin shell theory) and to the fact that the gross rather than the peak stress parameter is the object of this analysis; however the importance of this parameter is not large if $\rho / T \simeq 1.0$ (again from ref [3]).
In figs 2a, b,c, d, e, f the stress pattern in the ligament between the two nozzles along the meridian connecting the two nozzles centerlines is shown : the two components of the membrane and of the bending stress state are shown, as well as the largest stress intensity in the thickness.
In figs 3a, b, c, d one finds the stress intensity pattern along the nozzle circumference for various distances from the nozzle axis for the cases R150, R300, R400, R500.
In figs 4a, b the stress intensity pattern along the nozzles centerline axis both on the vessel and on the shell is shown for all the cases.

4. Discussion

The data are presented in detail in figs 2, 3, 4 however, for a discussion, it may be more useful a presentation of the data interms of stress intensity concentration factor defined as :

$$K_{SICF} = SI / SI_{\infty}$$

where :

SI = stress intensity

SI_{∞} = stress intensity where a full membrane stress exists

In fig 5a the K_{SICF} parameter is presented at various angles as a function of the parameter B_1 = d/D, while in fig 5b the stress concentration factor is presented directly at the center of the ligament connecting the two nozzles. As an indication, the S.C.F. by Seika et al. [3] has been also reported in fig 5a.
From the analysis of the fig one may derive the following conclusions :

a) the stress pattern around the nozzle is not sensibly influenced by the mutual interaction between the two nozzles even when the parameter B_2 is slightly less than the limit prescribed by ASME 3, this should hold at least for values of B_2 around 2.5.

b) the stress pattern at the center of the ligament is somewhat influenced by the parameter B_1, but the parameter B_2 is the most important in this case too

c) the stress concentration factor values, which have been obtained are of the same order of magnitude as reported in the open literature.

5. Conclusions

It is obviously impossible, due to the small number of runs performed by the authors, to draw general conclusions, besides this was not the aim of the paper. However the following tentative conclusions can be drawn :

a) the parameter $B_2 = \delta/d$ is the leading one in assessing the stress pattern mutual influence between the two nozzles, while some slight influence could be clained for the parameter $B_1 = d/D$

b) the limit $B_2 = 3.$, as presented by ASME is clearly a realistic somewhat conservative value to assess the mutual influence between the stress states of the two nozzles

c) even if B_2 is slightly less than 3. (say 2.5), no large influence of the nozzles coupling could be detected for the stress state around the nozzle, while the influence on the stress at the ligament center was much larger.

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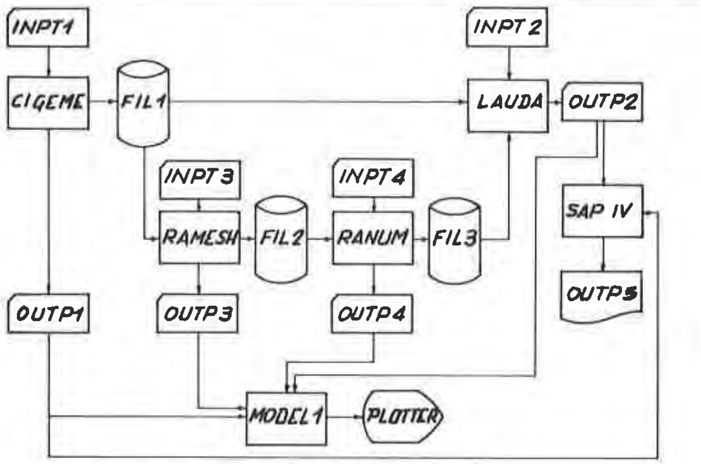

TAB 1

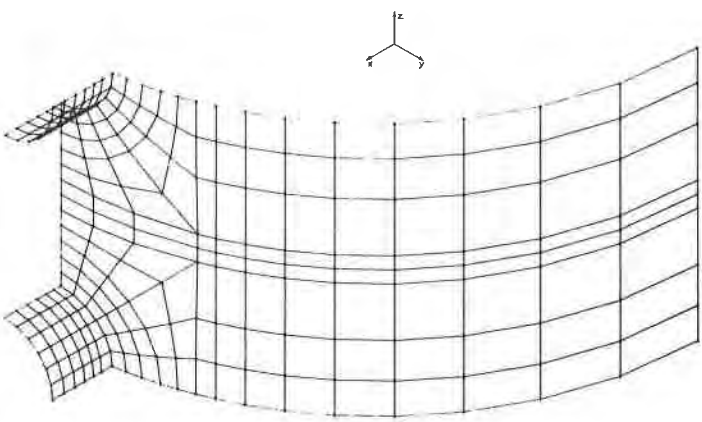

FIG. 1

STRESSES ALONG A LONGITUDINAL AXIS

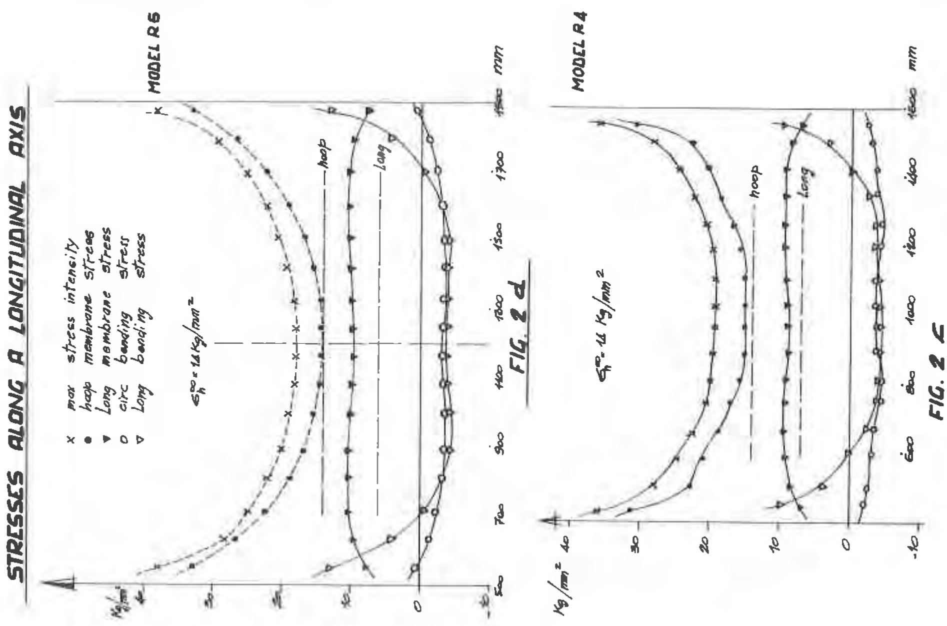

FIG. 2 d

FIG. 2 c

STRESS ALONG LONGITUDINAL AXIS

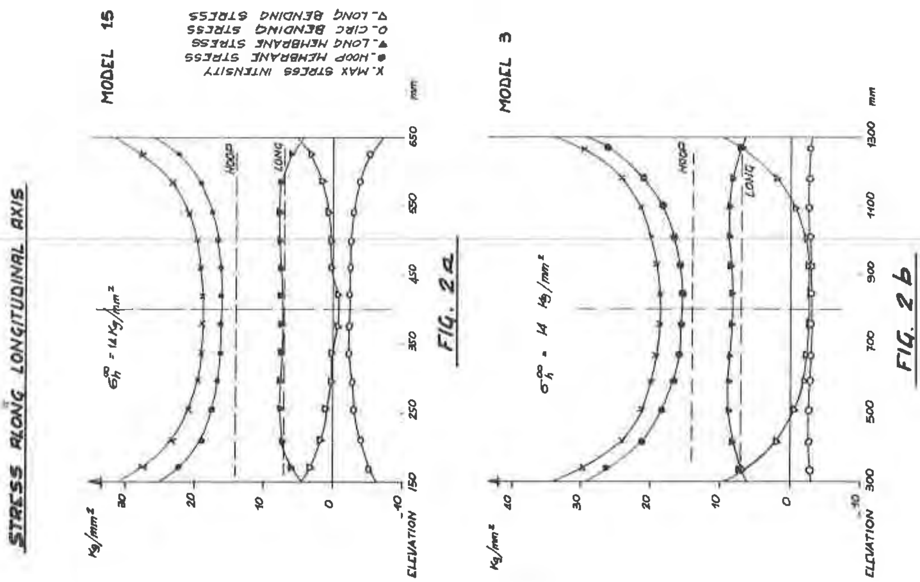

FIG. 2a

FIG. 2 b

STRESS ALONG LONGITUDINAL AXIS

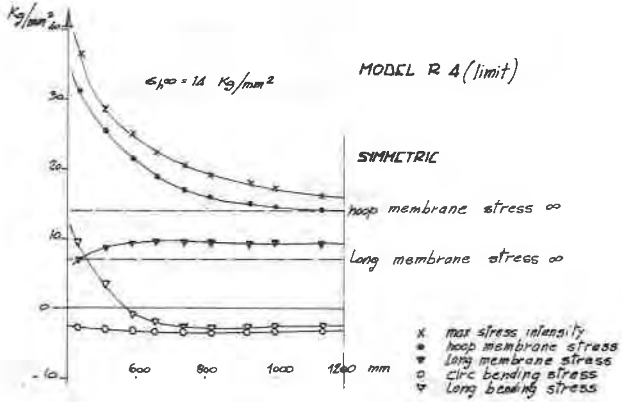

FIG. 2e

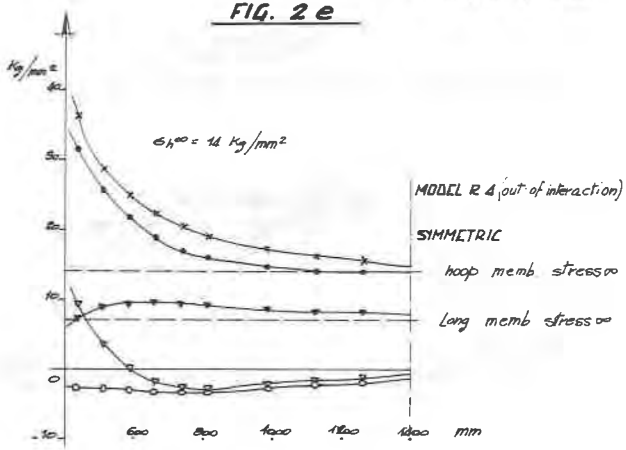

FIG. 2F

MAX STRESS INTENSITIES AROUND NOZZLE IN MAIN SHELL

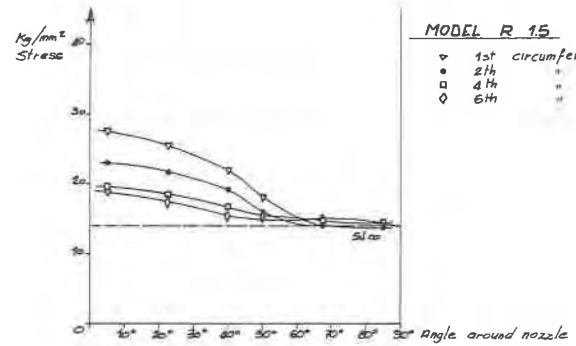

FIG. 3a

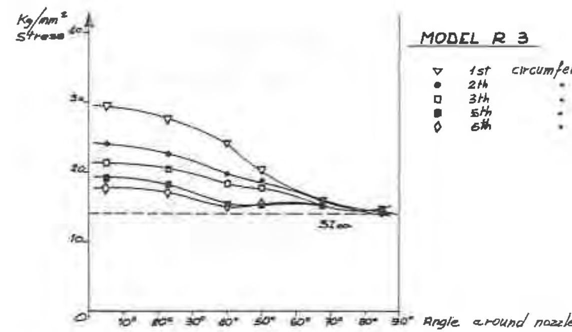

FIG. 3b

MAX STRESS INTENSITIES AROUND NOZZLE IN MAIN SHELL

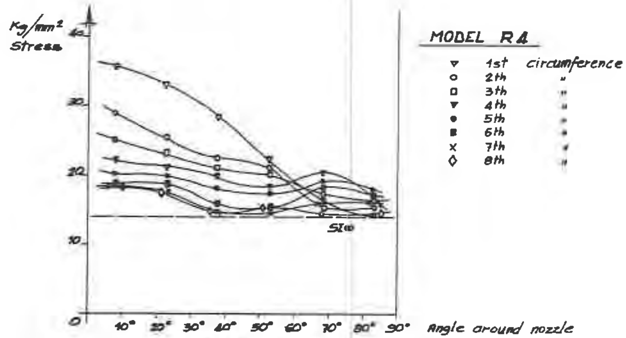

FIG. 3 c

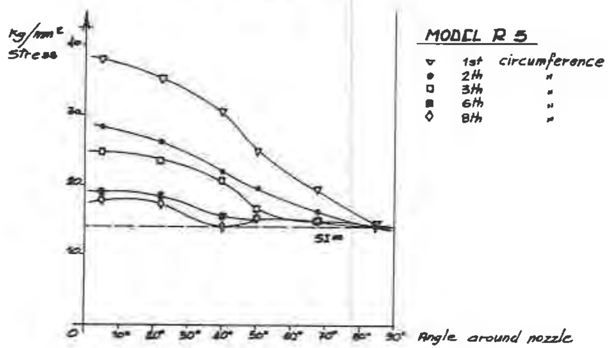

FIG. 3 d

MAX STRESS INTENSITIES AROUND NOZZLE IN MAIN SHELL

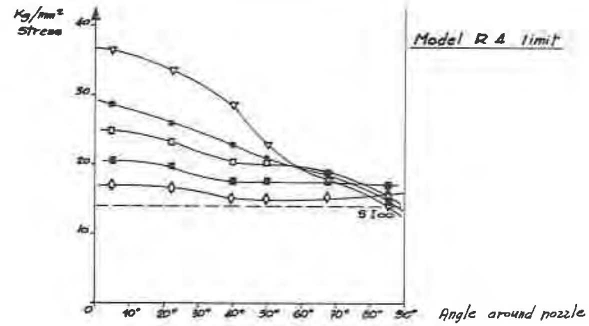

FIG. 3e

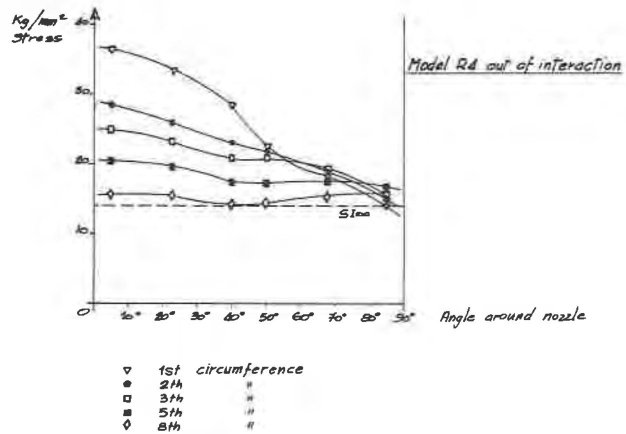

FIG. 3 f

MAX STRESS INTENSITIES

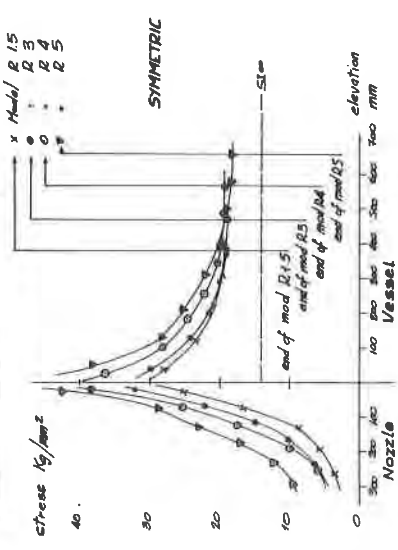

FIG. 4 a

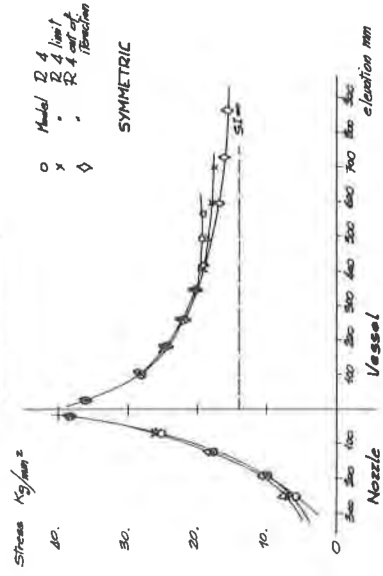

FIG. 4 b

STRESS INTENSITY FACTOR AT NOZZLE

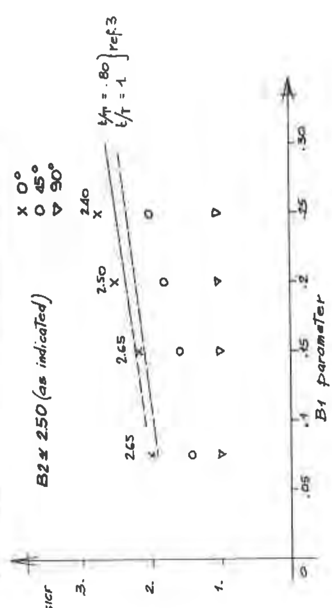

FIG. 5 a

STRESS INTENSITY FACTOR AT LIGAMENT CENTER LINE

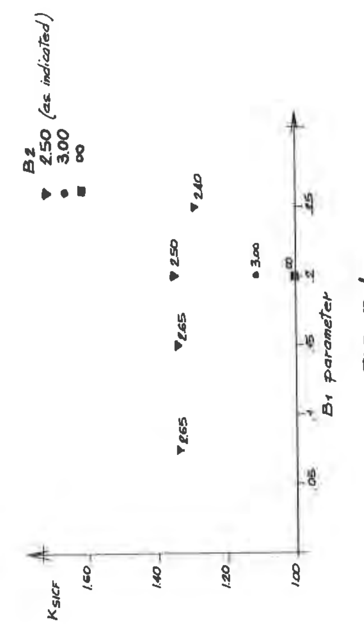

FIG. 5 b