THE FINITE ELEMENT STRESS ANALYSIS OF OBLIQUE PRESSURE VESSEL NOZZLES

by

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Contents

NOMENCLATURE

Unless otherwise indicated, the basic terminology applied in this thesis will be as shown below:

a	Semi-major axis of an ellipse
b	Semi-minor axis of an ellipse
d	Branch pipe mean diameters
D	Run pipe mean diameters
d_o	Branch pipe outside diameters
Do	Run pipe outside diameters
I_r	Run pipe second moment of area = $\pi/64(D_o^4-D_i^4)$
I_b	Branch pipe second moment of area = $\pi/64(d_o^4-d_i^4)$
M	Bending moment
P	Internal Pressure
r	Branch pipe mean radius
R	Run pipe mean radius
t	Branch pipe thickness
T	Run pipe thickness
b	Subscript-branch
r	Subscript-run
x,y,z	Space coordinates
σ_n	Nominal stress for internal pressure P = PD/2T (D= mean run pipe diameter)
σ_r	Nominal stress for bending moment applied to the run pipe = $MD_o/2I_r$
σ_b	Nominal stress for bending moment applied to the branch pipe = $Md_o/2I_b$

σ_{vm}	Effective stress based on Von Mises criterion
SCF	Stress concentration factor
ESF	Effective stress factor
FE	Finite element
FEA	Finite element analysis
FR	Fillet radius
deg.	degree
ASME	The American Society of Mechanical Engineers
BS	British Standards Institution
K_I	Stress concentration factor based upon the measured principal stress in question (at any given point) and the nominal stress in the vessel ($K_I=\sigma/S$); unless otherwise indicated, all references to "stress concentration factor" ("scf") are in terms of K_I.
K	Radius ratio R_o/R_i
fig.	Figure
Alpha	Skew-angle

"Inboard side"- the side of the connection closest to the vessel centerline for "hillside" connections in spheres and cylinders; also, the "acute" side.

"Outboard side"- the side of the connection farthest removed from the vessel centerline for "hillside" connections in spheres and cylinders; also, the "obtuse" side.

Analisis Unsur Tegasan Terhingga Untuk Bekas Tekanan Muncung Yang Berserong

ABSTRAK

Tegasan di bahagian 'inboard' bekas tekanan muncung yang berserong adalah sangat rumit dan tatacara rekabentuk untuk kes sebegini masih memerlukan penyiasatan. Walaupun demikian, setakat yang penulis tahu, tiada penyelesaian beranalisis yang sempurna wujud untuk meramalkan kesan terhadap tumpuan tegasan di bahagian inboard bekas tekanan muncung yang berserong. Kod-kod piawai ASME dan BS hanya memberi maklumat untuk bekas tekanan muncung yang berjejarian. Oleh yang demikian, penyelidikan ini bertujuan untuk memberi maklumat tentang bekas tekanan muncung yang berserong dengan menyiasat kesan pengubahan sudut serong, Alpha= 10°, 20°, 30° dan parameter yang tidak berdimensi seperti D/T(D/T= 10, 20), d/D(d/D= 0.20, 0.35, 0.50) serta nisbah t/T(t/T= d/D atau t/T≠ d/D) terhadap puncak tegasan di bahagian 'inboard' penyambungan bekas tekanan muncung.

Perisian Ideas Master Series 1.20 digunakan untuk menjalankan analisis tegasan unsur terhingga terhadap pelbagai model bekas tekanan. Keputusan penyelakuan dari bungkusan Ideas Master Series FEA itu dibandingkan dengan data tanda asas ujikaji (Moffat et al., 1989). Data keputusan penyelakuan sepadan dengan data ujikaji.

Dengan mengurangkan faktor tegasan berkesan (ESF), kita boleh menambah hayat lesu bekas tekanan muncung yang berserong. Dengan demikian, rekabentuk bekas tekanan muncung yang berserong boleh diperbaiki dengan merujuk kesannya terhadap ESF apabila dimensinya diubahsuai.

ABSTRACT

The stressing at the inboard side of the oblique pressure vessel nozzle is complex, and the design procedures in such cases are still in need of investigation. However, to the best of the author's knowledge, no complete analytical solution exists for predicting the effects on stress concentration at the inboard side of the oblique pressure vessel nozzle. Both ASME and BS codes only provide information for radial pressure vessel nozzles. Therefore, this research intends to provide information for oblique pressure vessel nozzle by investigating the effect of changing the skew angle, Alpha= 10°, 20°, 30°, and the dimensionless parameters such as D/T (D/T= 10, 20), d/D (d/D= 0.20, 0.35, 0.50), and t/T (t/T=d/D or t/T≠d/D) ratios on the peak stress at the inboard side of the juncture of the cylindrical pressure vessel nozzles.

Ideas Master Series 1.20 is used to carry out the Finite Element Stress Analysis on the various pressure vessel models. The simulated results from the Ideas Master Series FEA package are compared to the benchmark data from the literature (Moffat et al., 1989). The simulated results are in good agreement to the experimental data.

By reducing the Effective Stress Factor (ESF), the fatigue life of the oblique pressure vessel nozzles can be enhanced. Thus, the future design of the oblique pressure vessel nozzle can be improved by knowing what will be the consequences of ESF when the dimensions are being modified.

CHAPTER ONE

INTRODUCTION

1.1 GENERAL HISTORY

The finite element method is a powerful CAD tool for designing machine elements. In fact, the finite element method was initially created to be applied in the aircraft industry. Since then, this general numerical technique for the solution of partial differential equations subject to known boundary and initial conditions has undergone rapid development. Further, the development of the digital computer and the increasing capability of various modern technologies have enhanced the above numerical technique to become a powerful and versatile solution method for a huge range of advanced engineering problems.

1.2 GENERAL PROBLEMS AND SOLUTIONS

One of the critical problem about oblique pressure vessel nozzle is the failure at the juncture of the oblique pressure vessel nozzle. To solve this problem, the stress concentration factor (SCF) at the inboard side (the definition of inboard side can be found in fig. 2.7 and fig. 3.4) must be known in advance before designing the structure. However, it is no easy job to tackle this problem due to the tremendous cost of conducting experiments in the laboratory. In order to reduce the cost of conducting experiments and make the design in the practical way, the finite element analysis package, Ideas Master Series, can be applied to simulate several dozens of oblique pressure vessel nozzle models with different dimensions. The consequences of changing various dimensionless parameters on the ESF at the inboard side can be known. The accuracy of the simulation was validated by simulating the experimental models from the University of Liverpool, UK, and it was found that the simulated results were in good agreement with reference data.

1.3 GENERAL METHODS OF SOLVING THE ESF AT THE INBOARD SIDE OF THE OBLIQUE PRESSURE VESSEL NOZZLE

Basically, there are three ways of solving the high stress problem at the crotch corner/intersection of the pressure vessel nozzle. First, adding the material close to the opening. Second, shifting the material to form a specific shape. Third, shaping by metal removal from the vessel wall (Hassan, 1990).

Rather than shifting or adding the material at the inboard side/intersection, this research will change the dimensions (T, Alpha, D, t, d) to figure out that under what dimensions, the structure will have a low stress concentration at the inboard side/intersection. Also, it is good to mention here that this research is not intending to provide absolute solution for eliminating the high stress concentration at the above mentioned critical area. Rather, the research will provide a complete set of data to the pressure vessel engineers for predicting the changing of effective stress factor (ESF) when the parameters such as thickness, t, diameter, D, and etc. are changed. Also, the research will cover only three load cases, and these are internal pressure P, bending moments, M_{zr} and M_{zb}. The details of these loadings will be explained in the subsequent chapters.

1.4 OUTLINE OF THE THESIS

The second chapter of the thesis surveys the literature concerned with the oblique pressure vessel nozzle. The general problems of high stress concentration at the inboard side/intersection will be described in detail in this chapter. Apart from this, several formulae for calculating the SCF are also explained and presented in appendix A.

To generate the data for predicting the effects on ESF as dimensions of the oblique pressure vessel nozzle were altered, more than 80 models of this structure were created and simulated using the Ideas Master Series Finite Element Analysis Package. Also, 9 models of the Amran (Ayob, 1994) oblique pressure vessel nozzle with different angles were created to predict the effect on ESF as the skew angle was changed. The subject will be elaborated in more detail in chapter 4.

The fourth chapter brings the research to the high point as the significance of changing the parameters of the oblique pressure vessel nozzle will be discussed. Further, the simulated results of the Finite Element Analysis (FEA) will be compared to the Decock's experimental results (Moffat et al., 1991), and the discussion about this matter will be presented in this chapter.

To conclude the research, chapter 5 brings the overall discussion and recommendation from chapter 4 to the final stage, and the conclusions of the research are listed in this chapter.

Chapter Two

Literature Review

2.1 Introduction

To the best of the author's knowledge, no complete analytical solution exists for predicting the effects on stress concentration at the inboard side (the location of inboard side can be seen in fig. 2.3, fig. 2.7 and fig. 3.4) of the oblique/hillside (the illustration of hillside pressure vessel nozzle can be seen in fig. 2.6) pressure vessel nozzle. Nevertheless, this is a kind of high risk engineering, and thus, it receives enormous amount of attention from the pressure vessel and piping engineers. Pressure vessels are containers with leakproof function. However, the design of the pressure vessel nozzle is complex. Pressure vessel nozzle intersections occur frequently in pressure vessel nozzle systems at locations where flow is required to diverge or converge. The sudden change of direction and thickness at the intersection of the pressure vessel nozzle cause high stress concentration at the abrupt change point, and thus, it creates a high possibility of cracking.

The stressing of the pressure vessel nozzle intersections is complex due to the geometric variables involved, the method of manufacture, the various possible load categories, and the design criteria to be fulfilled. It is generally recognized that there are seven basic load categories which apply to pressure vessel nozzle junctions. These are internal pressure P and six moment loads depicted in fig. 2.1, using the so-called cantilever model (Hassan, 1990). Moment loads can arise as a result of combined deadweight, thermal expansion, cold-pull, and dynamic effect. All seven loads frequently occur together, and each has its own complex stress distribution (Moffat et al., 1989). To predict the effects on stress concentration at the inboard side of the

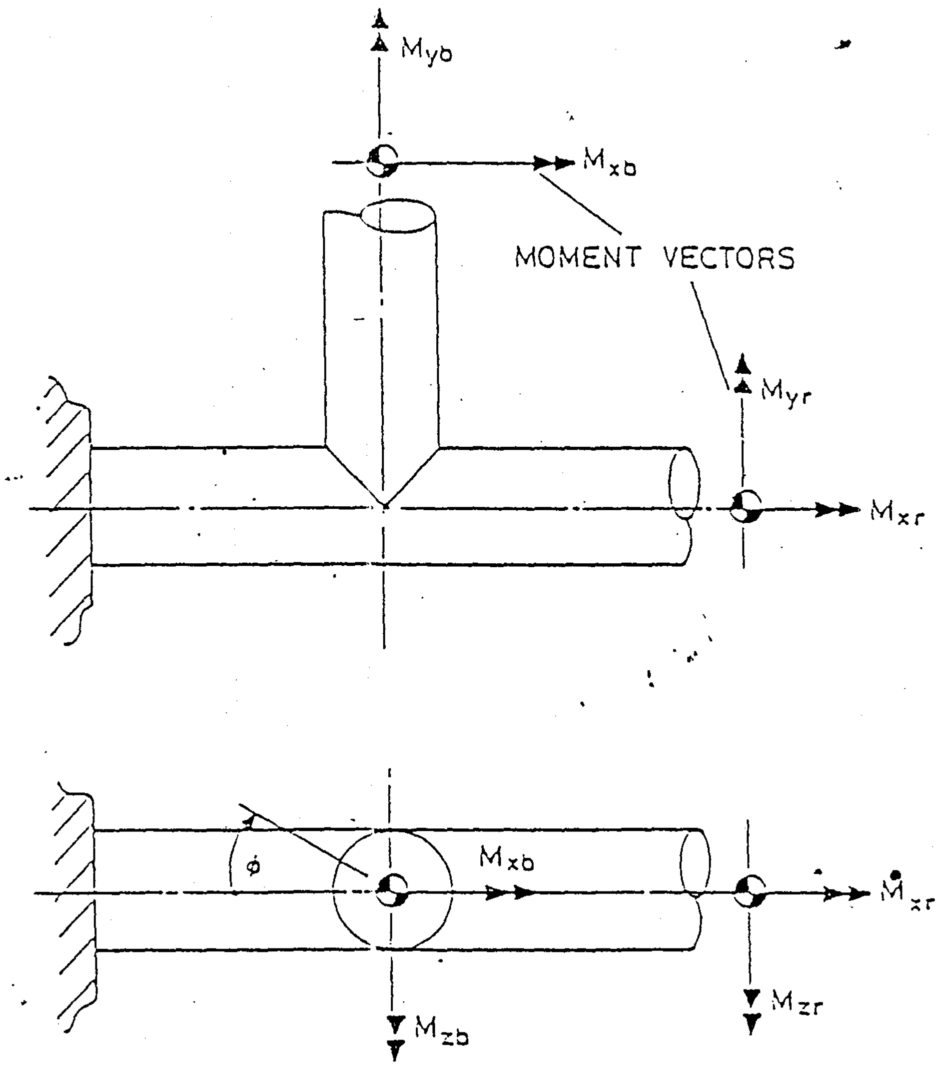


Fig. 2.1 Notations for Bending Moment Loadings (Hassan, 1990)

oblique pressure vessel nozzle, numerical solution on this matter was done by applying three types of loads separately on the pressure vessel nozzle in this research. These are internal pressure P and bending moments, M_{zr} and M_{zb}.

2.2 Oblique Pressure Vessel Nozzle

To build an oblique pressure vessel nozzle, a hole must be drilled on the pressure vessel to allow the access for both the flowing fluid and personnel. As a result, welding must be made to compensate the "loss material". So that the new-formed structure can withstand the internal pressure and external bending moment.

The joining of nozzle and pressure vessel at the "hillside" of the pressure vessel causes serious stress concentration at the inboard side/intersection at pressure vessel and nozzle. The high stress concentration over these regions create serious potential failure.

In the past, the method of preventing the junction of the pressure vessel nozzle from collapse is to reserve a large strength to withstand the hypothetical over-pressure conditions. However, this method will not be applicable if the leakage is due to the crack at the local peak stress area. Therefore, it appears that the distribution of stress should be modified to the extent that each point will have an equal stress on the oblique pressure vessel nozzle. On the other hand, it will be almost impossible to distribute stress to each point equally due to the complexity of the structure. Thus, this research does not aim for the total elimination of high stress concentration at the intersection. This research serves the purpose of providing new information about the effects of varying dimensions of the structure on high stress concentration at the inboard side.

2.3 Stress Concentrations

The fatigue crack growth and established fatigue life are the functions of the magnitude of the applied stress. Therefore, if the "stress raiser" is created on the nozzle and pressure vessel then the life of the structures will be affected due to the direct multipliers of the normal stress. Geometric discontinuities such as holes, grooves, notches, abrupt changes in cross section, etc., as well as thermal discontinuities, cause a local increase in stress, and proper allowance for the effects of these stress concentrations is the single most important factor in designing to resist failure by fatigue (Harvey,1985). In short, the control and the way to predict the stress concentration at the inboard side/intersection will be very significant in increasing the service life of the oblique pressure vessel nozzle.

2.4 Localized stresses and their significance

The ordinary formulae that have been created for the stresses in pressure vessels are based on the assumption that there is continuous elastic action throughout the member and the stress is distributed on any cross section of the member by a mathematical law. For example, in the case of simple tension and compression, it was presumed that the stress was uniformly distributed over the whole cross section. As these basic assumptions are no longer valid due to the abrupt changes in section or part, tremendous irregularities in stress distribution happen which, in general, are established only in a small section of the member, and thus, they are called localized stresses or stress concentrations. In pressure vessels, they happen at the intersections at thick and thin sections of the shell, and about openings, nozzles, or other attachments.

In approaching the study of localized stresses, it is well to note that their significance does not depend solely on their absolute value. It also depends on:

1.) The general physical properties of the material such as its ductility.

2.) The relative proportion of the member highly stressed to that under stressed which affects the reserve strength it can develop in resisting excessive loads.

3.) The kind of loading to which the member is subjected; i.e., whether it is static or repeated loadings.

For instance, stress concentrations in a pressure vessel subjected to only a steady pressure, or one repeated for only a few times, are of little significance provided the vessel is made of ductile material, such as mild steel, which yields at these highly stressed locations allowing the stress to be transferred from overstressed fibers to adjacent understressed ones. Riveted boiler drums are a good example of this. Localized stresses in the portion of the rivet holes achieve yield point values, yet failure does not happen since they are made of a ductile material and operated at substantially steady-state conditions.

However, if the loading is a repetitive one, localized stresses become important even though the material is ductile and has a great measure of static reserve strength. Likewise, the stresses given by the ordinary formulae which are based on average stress conditions, and which do not consider for such local effects, must be multiplied by a theoretical "stress concentration factor, K_I," defined as the ratio of the maximum stress, to get the maximum stress value (Harvey, 1985).

2.5 Determination of high local stresses

The mathematical approach of these localized stresses is usually impossible or unrealistic, thus, experimental approach of stress analysis are usually applied (Harvey,1985).

To overcome the difficulties of the analysis job, Finite Element Analysis (FEA) is applied to compute the SCF of certain cases as mentioned above (Harvey, 1985). The bulk of information available concerning with high localized stresses is the results of analytical and experimental investigations of thin-walled sections. Due to the complexity of the configuration of the typical nozzle-vessel intersection, classical mathematical approaches of stress analysis are not realistically applicable. Thus, finite element stress analysis (FEA) method is a realistic approach for such a problem. Application of the finite element method is generally for vessel with thinner wall thickness. The simulated results are in good agreement with the experimental results (Hassan, 1990). In this research, D/T ratios for most of the models are between 10 to 20 which generally can be considered as thin wall vessel; according to BS 5500 (University of Strathclyde, 1989), the pressure vessel is thin in this sense if D/T ≥10. Also, the complexity of intersection at the pressure vessel and nozzle increases due to the weld is added to the juncture at nozzle and vessel. Therefore, FEA has to be applied to do the analysis.

Some form of axisymmetric approximation is usually applied to achieve a suitable SCF for a nozzle in a cylindrical vessel. Indeed, researchers even apply a cylinder/sphere model in which the sphere has twice the diameter of the real vessel. The stresses calculated at the intersections are taken as indicative of those in the real geometry. Hassan (1990) mentioned that Truit and Raju conducted a comparative study between 3-D and an axisymmetric finite element analysis of a reactor pressure vessel inlet nozzle subjected to internal pressure. This research proves that the axisymmetric analysis is unconservative if based upon common modeling techniques.

2.6 Design approach

The design of most structures is based on formulae that are known to be approximate. The unknown items, such as extent of yielding and the omitted factors in design and material behavior, are considered to be provided for by the use of working stresses that are admittedly below those at which the member will fail. This factor of safety process, even though contains virtue of having worked effectively in the past for ductile materials under static loading and providing the designer with preliminary sizing data, is producing to more refined analytical and experimental approaches. In fact, factor of safety are a trade-off means of establishing equal performance credibility and safety by consigning to a single parameter varying degrees of quality assurance (design analysis, material testing, fabrication control, and inservice inspection). This progress will carry on as knowledge and cognizance of influencing design and material parameters increase are put to engineering and economic utilization. This might be described by the triangle of knowledge as in fig. 2.2, which indicates that as our ignorance reduces with discovery or recognition of additional variables affecting behavior and proper consideration is taken in design analysis, the chances for error reduces; accordingly, the potential properties of the substance can be more completely applied with confidence.

The pressure vessel material behavior and stress analysis knowledge have been accelerated by the safety requirements of nuclear reactors, deep-diving submersibles, space vehicles and chemical retorts.

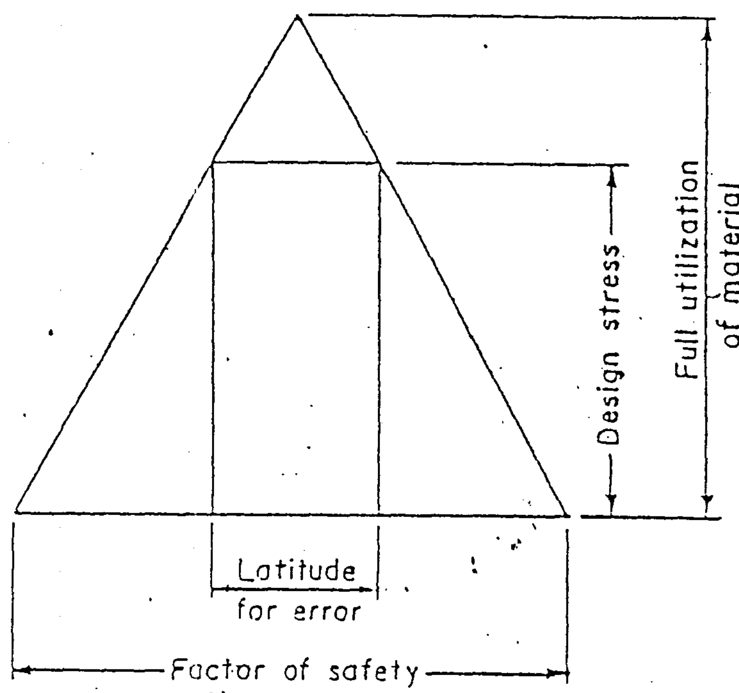


Fig. 2.2 Triangle of knowledge (Harvey, 1985)

High strength materials, created by alloying elements, manufacturing processes or heat treatments, are developed to satisfy economic or engineering demands, such as reduced vessel thickness. They are continually being tested to establish design limits consistent with their higher strength, and adapted to vessel design as experimental and fabrication knowledge justifies their application. There is no one perfect pressure vessel material suitable for all environments, but material selection must match use and environment. This has become especially significant in chemical reactors because of the embrittlement effects of gaseous absorption, and in nuclear reactors because of the irradiation damage from neutron bombardment.

Major progresses, extensions and developments in analytical and experimental stress analysis are allowing fuller utilization of material properties with confidence and justification. Many formerly insoluble equations of elasticity are now yielding to computer adaptable solutions, such as the finite element method and these together with new experimental techniques have made possible the stress analysis of structural discontinuities at nozzle openings and attachments. This is important because 80% of all pressure vessel failures are caused by the high localized stresses associated with these "weak link" construction details. Thus, stress concentrations at the vessel nozzle openings, attachments and weldment are of prime importance, and approaches for reducing them through better designs and analyses are the keys to enhance the pressure vessel life (Harvey, 1985).

2.7 Design of Nozzles in Pressure Vessels

To fulfill the safety specifications for a pressure vessel, the following possible modes of failure have to be accounted as design basis (Hassan, 1990):

* excessive elastic deformation

* excessive local plastic deformation

* fatigue failure caused by high local stresses often combined with peak stresses at discontinuities

* excessive overall plastic deformation of plastic instability

* brittle fracture, corrosion and so on, largely depending on the material properties but also associated with high local or peak stresses.

It is the current method for the design proposals to be according to the results of extensive analyses by computer with comprehensive experimental proof. Hassan (1990) indicated that Moore, Greenstreet, and Mershon provided a short discussion on modern pressure vessel design philosophy with a background review for the design analysis approaches. Comments were made that in most cases, elastic stress analysis approaches supplemented by general understanding of plasticity are suitable. In certain cases, however, solutions obtained directly from plasticity and creep theories are applied; and in more sophisticated research, plasticity and creep are becoming more significant.

Gill (Hassan, 1990) mentioned some views made by Leckie and Payne that if a pressure vessel is simply subjected to a constant pressure outside the creep range, then limit analysis would be the suitable design basis. If high strain fatigue or incremental collapse are to be avoided for vessels subjected to cyclic pressure, then the structure should satisfy the shakedown criterion at the design pressure. Where fatigue failure is possible with a high number of pressure cycles, the maximum elastic stress concentration factor should be the design criterion.

2.8 Dimension Parameters of Nozzle Vessel-Intersections.

The diameter ratio d/D of the nozzle to the vessel is commonly less than 0.5 in pressure vessel. In piping, the d/D range is usually from 0.5 to 1.0. The difference between nozzles in pressure vessels and branch connections in piping is only in the application. The parameter that differentiate the thick and thin cylinders is the D/T ratios. High ratios are categorized as thin, and low ratios are identified as thick. The stress analysis for the cylinder to cylinder junctions often based on thin shell theory and the requirement for its application is that ratio diameter to thickness for both cylinder and nozzle must be more than or equal to 20 (Hassan, 1990).

2.9 The Effect of Ellipse Ratios (a/b) on the Stresses at the Nozzle-Vessel Intersections Under Internal Pressure Loading

Hassan (1990) has conducted research on the effect of ellipse ratios (a/b) on the stresses at the intersection of the pressure vessel nozzle under internal pressure loading. A significant stress reduction was obtained when the intersection were elliptically shaped and the extent of the SCF reduction based on the experimental work is given in table 2.1.

Table 2.1 SCF Reduction at Crotch Corner and Maximum SCF Reduction in the Crotch Region (Hassan,1990)
d/D =0.2 D/t ≈8 K=1.287

Ellipse Ratios (a/b)	SCF at crotch	SCF % SCF Reduction	SCF maximum value	SCF % SCF Reduction
1.0	3.24	0	3.24	0
2.0	2.24	30.9	2.24	30.9
2.66	1.90	41.4	2.05	36.7
3.20	1.70	47.5	1.80	46

The above results will be compared to the results in table 3.13 in chapter 3, and the complete discussion about this subject will be elaborated in more detail in chapter 4.

2.10 Methods of SCF (ESF) Reduction in An Oblique Pressure Vessel Nozzle Intersection.

In general, very little information was available about the above mentioned subject until recently because either it was not published or the company conducting the experiment felt that the information was proprietary. Nevertheless, within the past few years, a number of papers and reports have been published providing data for specific problems concerning nonradial nozzles (WRC Bulletin, 1970).

Basically, the stress-raising effect of a nozzle depends mainly on the thickness diameter (D/T) ratio of the vessel and the diameter ratio of the nozzle to the vessel, d/D. There are three possible methods of reducing the SCFs near the nozzle-vessel juncture (Hassan, 1990).:

1.) Adding material close to the opening (area replacement);

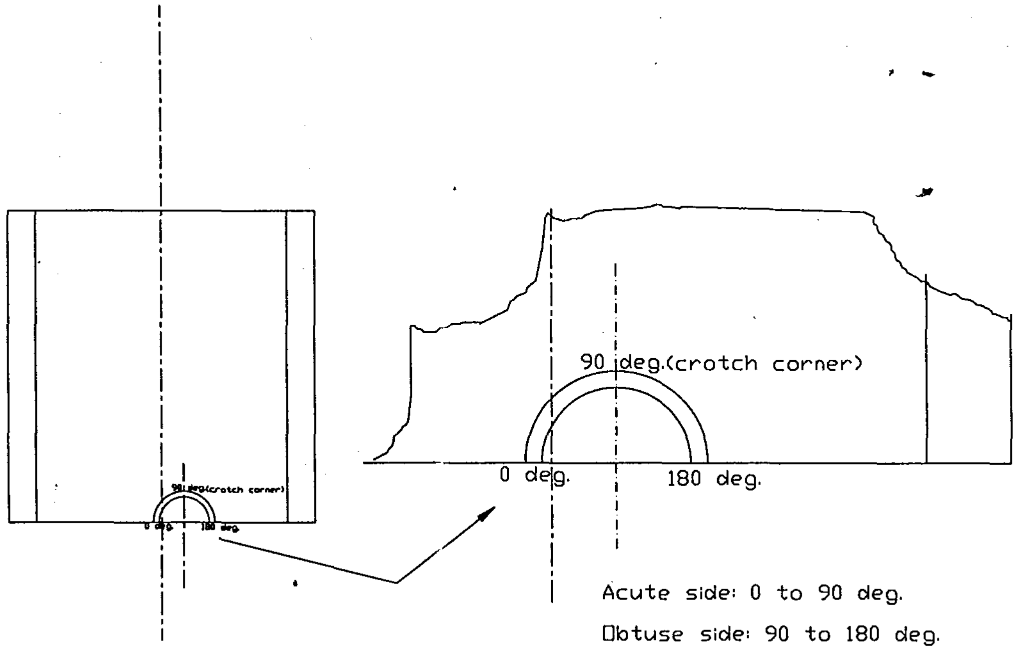
2.) Shifting material to form a specific shape (forming eg. forging, extrusion);

3.) Shaping by metal removal from the vessel wall (elliptical nozzle-vessel juncture)- this method is only relevant to thick walled vessels.

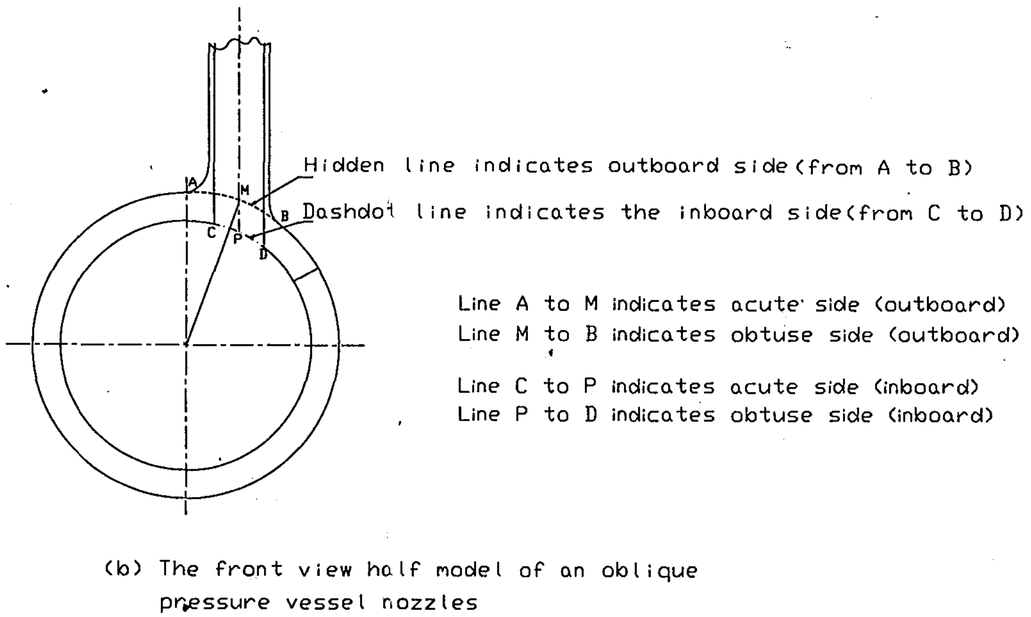
2.11 "Hillside" Nozzle Connections on a Cylinder

A "hillside" nozzle (the illustration of hillside nozzle can be seen in fig. 2.6) connection on a cylinder is generally assumed to be a less serious design problem than a lateral connection (the illustration of lateral connection can be seen in fig. 2.6) on a cylinder or a "hillside" connection on a sphere because the elliptical cut-out is oriented favorably with respect to the 2:1 biaxial stress field in the shell. Nevertheless, since the stresses in the "acute corners"(the location of acute corner can be seen in fig. 2.3) had proved critical in prior tests, an exploratory test was made on one oblique nozzle where the "corner effect" would presumably be a maximum; also, such a connection might-in some measure-provide qualitative guidance concerning critically stressed regions in a pump outlet connection. Subsequently, it was learned that American investigators had conducted a series of photoelastic tests on connections of this type, the results as shown in Table 2.2 (Mershon,1970). From these data the following points may be noted:

(a) Fidler's data indicate that the effect of increasing obliquity on the stresses, measured in the bore of the nozzle on both the major and minor axes of the connection, are basically similar to that for the case of an elliptical hole in a flat plate under the same state of stress. The agreement is particularly good for the stress on the longitudinal axis of the shell (equal to the minor axis of the hole), as shown on Fig. 2.4a, although, the stresses for the radial nozzle (F-1A) appear "high" in relation to other available data-both photoelastic and steel model F-1C at Alpha= 50° are slightly "low" in relation to those from the remaining models. As has been demonstrated by the authors' own experience, occasional "scatter" in the data is seemingly inevitable. The same basic trend seems to apply to the stresses on the minor axis of the Allison-Hay models and of Westinghouse models WC-12C1 and WC-12C3. A considerably flatter trend curve is indicated by the Foster-Wheeler model data; however, these data were from an "as fabricated" steel model in which out-of-roundness effects were quite apparent at some locations in the shell and this could have affected the measured stresses significantly (Mershon, 1970).



(a) Top view of a typical
oblique pressure vessel nozzle



(b) The front view half model of an oblique
pressure vessel nozzles

Fig. 2.3 The definitions of acute side-inboard,
obtuse side-inboard, acute side-outboard,
and obtuse side-outboard of an oblique pressure
vessel nozzle

In the case of the stresses on the major axis of the hole, there are seeming anomalies in the data which are only partially explainable. The following points might be noted in examining the data (Mershon, 1970):

(1) The maximum stresses on both the major and minor axis of the two Westinghouse models WC-12C1 and WC-12C3 are essentially identical except for a relatively minor difference on the inboard side in the bore of the nozzle (SCF of 1.25 vs. 1.50). The maximum stresses in the bore of the nozzle on both the inboard and outboard sides are significantly higher than for the radial counterpart, model C-2A (counterpart of model WC-12C1).

(2) Fidler's data on the major axis shows an increasing trend on both the inboard and outboard sides with increasing angle of obliquity. However, in comparing the stresses in model F-1D with its approximate, unreinforced counterpart, Westinghouse model WC-12C3, it will be noted that the "critical" stress in model F-1D is at the inner, acute corner, whereas the critical stress in WC-12C3 is at the base of the nozzle on the outboard side. However, this difference can be explained that the addition of outward rotation thrust will cause an outward rotation at the edge of the hole on the transverse or minor axis. This in turn will create bending stresses at the ends of the major axis, which are positive (tensile) on the outside surface and negative (compressive) on the inside surface. The addition of a nozzle will restrain this outward rotation; as the thickness of the nozzle is increased, the circumferential stresses, σ_n, will be gradually reduced. However, on the inside surface, the initial effect will be to increase the circumferential stresses as the bending component of stress (outward rotation) is eliminated; thereafter, with further increases in nozzle wall thickness, the stress will be reduced somewhat, as the reinforcement begins to "control" the membrane stresses also.

(3) The stresses in the Allison-Hay model, AH-1B, are very similar to those in Fidler's model F-1B (at a comparable angle of obliquity, but differing in diameter ratio) with one notable difference-namely, the critical stress in model AH-1B is the tangential stress on the outer surface of the inboard side of the connection. It appears probable that the higher stress in the latter case is due to: (1) a tendency for the acute angle to "open up" in large size connections. (2) a relatively small fillet radius at the toe of the triangular weld fillet (r/T=0.14). A similar tendency might be expected in comparing Westinghouse models WC-12C1 and WC-12C3. In this case, the tangential stress, σ_t, for model WC-12C1 is perhaps higher than for model WC-12C3, although not markedly so (noting that the reported stress for model WC-12C1 involves an extrapolation which is somewhat uncertain). Presently available data are not sufficient to define the parametric conditions under which the stress at this location becomes "critical."

(4) In spite of some uncertainties in the data, it will be observed that the effect of increasing obliquity on the stresses in the bore of the nozzle on the major axis of the hole follows the *qualitative trend* for elliptical holes in a flat plate, as shown in fig. 2.4b.

(b) For very small holes, the results from two models, models WC-12C and WC-12C2 with d=T/4, and Alpha= 45° and 78°17′, respectively, show that the stresses are appreciably less than the 2.5 factor, particularly for the latter model. Nevertheless, the plotted results, shown on fig. 2.4a and fig. 2.4b, give trend curves which are considerably "flatter" than for the larger nozzle connections, tending toward the radial hole values (scf =2.5 on the minor axis and 0.5 on the major axis). Further, for angles of obliquity between approximately 40° to 65°, it appears that the maximum stress in small holes may be slightly higher than those for larger connections. The practical

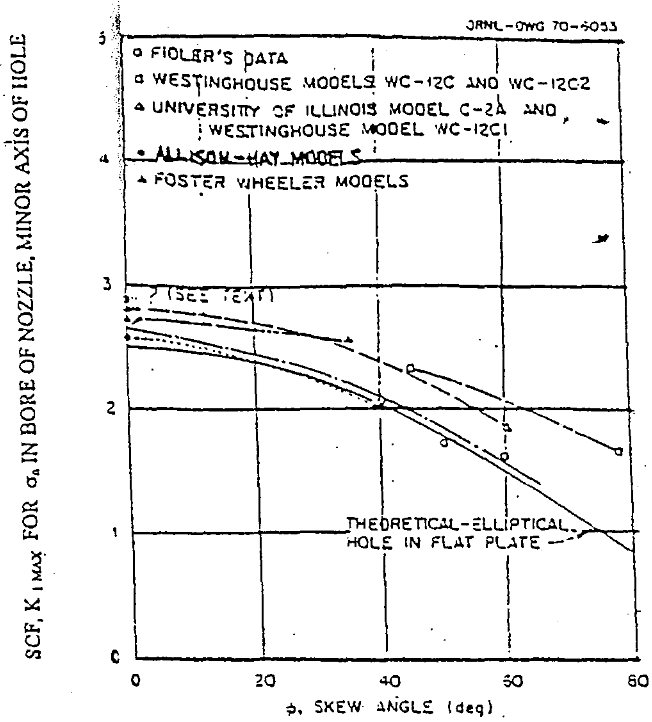


Fig. 2.4a Effect of skew angle on stresses in a "hillside" connection on a cylinder (longitudinal axis of vessel-minor axis of hole) [Mershon, 1970]

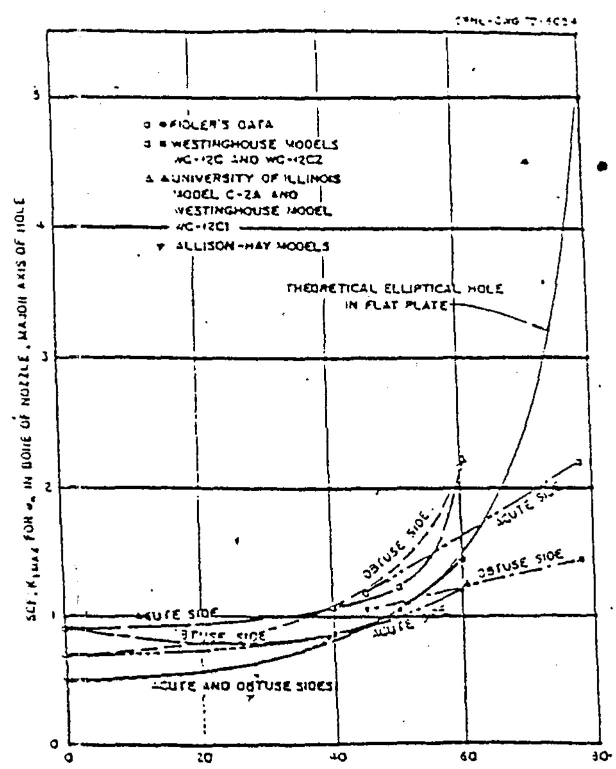


Fig. 2.4b Effect of skew angle on stresses in a "hillside" connection on a cylinder (transverse axis of vessel-major axis of hole) [Mershon, 1970]

significance of this stress in terms of comparative fatigue performance, etc., is uncertain (Mershon, 1970).

2.12 Steel Model Fatigue Tests

Mershon (1970) mentioned that Heirman, Stockman and co-workers also tested a number of "hillside" connections on a cylinder. Unfortunately, there are certain inconsistencies in the results, particularly in the latter series of tests, which are believed primarily attributable to poor fit-up and to the quality and finish of welds. Nevertheless, for 10 *flush* models tested, 9 out of 10 failed on the inboard side of the connection, starting from the toe of the weld fillet on the outboard side. It was also noted that for the largest d/D ratio, at the higher angle of obliquity (model HS-E in Table 2.2), there was a noticeable tendency for the connection to "open up" on the inboard side.

These results indicate that the inboard side of the connection is probably the critically stressed area for "hillside" connections in all but the smallest sizes. Since this stress is partially a bending stress at the edge of the connection, its actual magnitude will vary with D/T ratio, t/T ratio, angle of obliquity and the nature of the transitions used between the nozzle and the shell. For fairly "typical" nozzle connections having d/D ratios between, say, 0.10 and 0.25, this would appear to become the critically stresses area at angles of obliquity in the order of 30° or more. For lesser angles of obliquity, the maximum stress will ordinarily be on the longitudinal axis of the vessel (the same as for a radial nozzle).

Table 2.2 Stresses in "Hillside" Nozzle Connections on a Cylinder (Mershon, 1970)

Model	D_i/T	d_i/D_i	d_i/T	t/T	s/S	r_i/T	r_o/T	φ	Max. scf, K_t, in nozzle bore: Longitud. axis	Acute side	Obtuse side	σ_t max. outside, acute side
WC-12C	12.2	0.020	0.241	0	—	0	0	45	2.33	1.17	1.04	—
WC-12C2	12.2	0.022	0.25	0	—	0	0	78.3	1.67	2.20	1.43	—
C-2A	12.1	0.129	1.56	0.133	0.973	0.556	0.556	0	2.80	~0.70	~0.70	1.01
WC-12C1	12.2	0.129	1.58	0.127	1.01	0	0.32	60.6	1.87	1.25	2.23	1.45
WC-12C3	12.2	0.067	0.80	0.067	0.994	0	0.36	60	1.87	1.50	2.21	1.29
AH-1A	13.3	0.162	3.9	0.556	0.343	0	0.185	0	2.56	0.49[b]	0.49[b]	1.70
AH-1B	13.3	0.162	3.9	0.556	0.343	0	—[a]	39.3	2.02	0.83[b]	0.82	2.18
F-1A	15.0	0.067	1.00	0.242	0.321	0	—	0	2.86	0.90	0.90	1.29
F-1B	15.0	0.067	1.00	0.242	0.321	0	—	40	2.07	1.06	0.87	~1.1
F-1C	15.0	0.067	1.00	0.242	0.321	0	—	50	1.73	1.22	1.05	~1.1
F-1D	15.0	0.067	1.00	0.242	0.321	0	—	60	1.62	2.16	1.45	~1.25
FW-15	10.85	0.163	1.78	0.142	1.05	0.03	—	0	2.72[c]	—	—	—
FW-17	10.85	0.163	1.78	0.142	1.05	0.03	—	35	2.56[c]	—	—	—
SWRI-2B	18.0	0.266	4.78	1.20	0.261	0	—	0	2.92	—	—	—
HS-A	27.5	0.387	10.45	1.62	0.267	0	—	0	2.6[d]	—	—	—
HS-D	27.5	0.387	10.45	1.62	0.267	0	—	22.7	—	—	—	3.05[f]
HS-E	27.5	0.387	10.45	1.62	0.267	0	—	34.5	1.2[e]	—	—	3.4[f]

[a] Triangular fillet with $0.063T$ radius at toe of fillet.
[b] Measured on inside shell surface immediately adjacent to tip of corner.
[c] Based on calculated mean diameter stress. See Paragraph 6(a) of text.
[d] Measured $0.45T$ from corner; probably not maximum.
[e] Measured $\sim 1.0T$ from corner; almost certainly not maximum.
[f] Estimated value based on (2.92 × fatigue str. reduction factor); see text.

2.13 Moment loads

To the best of the author's knowledge, not much information is available on the oblique pressure vessel nozzle under moment loads. However, some information about radial pressure vessel nozzles under moment load are available. Moffat et al. (1991) have done some researches on radial pressure vessel nozzle under moment loads, M_{zr} and M_{xb} . Fig. 2.5a and 2.5b show the ESF results for loads M_{zr} (in-plane run moment) and M_{xb} (out-of-plane branch moment), respectively. It is encouraging that the various manually-fitted curves through the FE data in the range $0.2 \leq d/D \leq 0.8$ extrapolated back quite nicely to the deduced values for very small nozzles. The FE data match up well on the whole with the experimental data for d/D= 1.0. These latter data were determined using the empirical equations of the form:

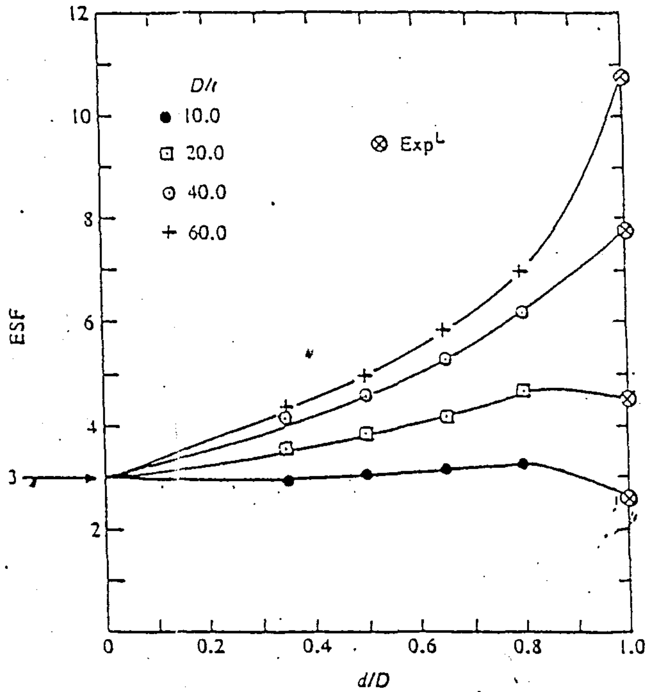
$$ESF = Q(R/T)^n$$

where, for the six moment loads, Q and n are given in table 2.3 as shown below:

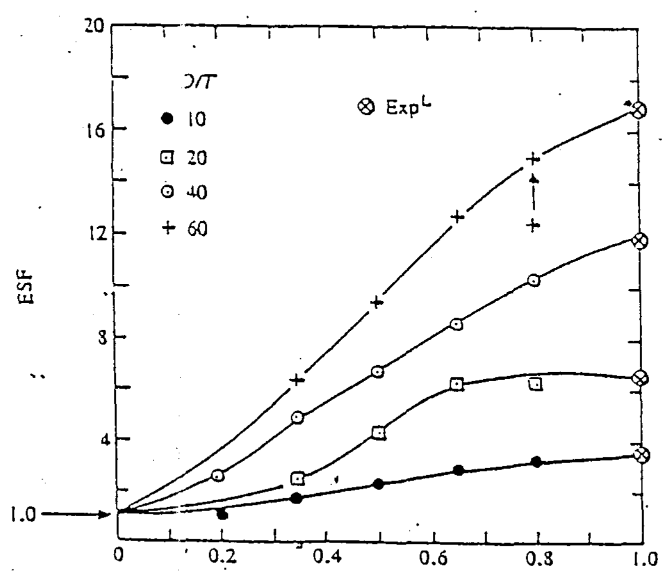
Table 2.3 Data for empirical relationships for ESFs where d/D= 1.0 (Moffat et al., 1991)

Moment load	M_{xr}	M_{yr}	M_{zr}	M_{xb}	M_{yb}	M_{zb}
Q	0.61	0.78	0.73	0.88	1.25	0.90
n	0.94	0.47	0.79	0.87	0.71	0.78

The ESF for Model 80D (d/D= 0.8, D/T= 60) for M_{xb} loading (fig. 2.5b is below the curve drawn through the other points). This case was re-run and the result confirmed. However, it is recommended that an ESF of 15.0 be assumed for this case, rather than the calculated value of 12.5. This is conservative and will assist in the curve fitting exercise to follow.



(a) Run pipe in-plane moment, M_{zr}



(b) Branch pipe out-of-plane moment, M_{yb}

Fig. 2.5 Effective stress factors for moment loads M_{zr} and M_{yb} with $d/D = t/T$ (Moffat et al., 1991)