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Systematic Design of Cylinder Heads for Reciprocating Compressors

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ABSTRACT

The design of reciprocating compressor cylinder heads consists of three main tasks: 1) calculation methods to determine the actual stresses, 2) determination of the factors of safety given these stresses, 3) validation of these stresses with strain gage data. This paper will address the first of these tasks, namely, the calculation of cylinder head stresses. A generalized procedure calculating stresses in reciprocating compressor cylinder heads, independent of FEA, is described. This method can be used for the preliminary design method prior to a costly FEA. It can also be used as a design tool for designers that may not have access to FEA software.

NOMENCLATURE

M_{1A}	Internal moment on element "A" at juncture 1, figure 2.
Q_{1A}	Internal shear force on element "A" at juncture 1, figure 2.
M_{1B}	Internal moment on element "B" at juncture 1, figure 2.
Q_{1B}	Internal shear force on element "B" at juncture 1, figure 2.
R_{BOLT}	External bolt force.
P_{GASKET}	External gasket pressure.
M_{2B}	Internal moment on element "B" at juncture 2, figure 2.
Q_{2B}	Internal shear force on element "B" at juncture 2, figure 2.
M_{2C}	Internal moment on element "C" at juncture 2, figure 2.
Q_{2C}	Internal shear force on element "C" at juncture 2, figure 2.
P_{GAS}	External gas pressure.
F_{S1A}	Internal axial load on element "A" at juncture 1, figure 2.
F_{S1B}	Internal axial load on element "B" at juncture 1, figure 2.
F_{S2B}	Internal axial load on element "B" at juncture 2, figure 2.
F_{S2C}	Internal axial load on element "C" at juncture 2, figure 2.
Θ_{M1A}	Rotation of element "A" at juncture 1, due to M_{1A}.
w_{M1A}	Radial displacement of element "A" at juncture 1, due to M_{1A}.
Θ_{Q1A}	Rotation of element "A" at juncture 1, due to Q_{1A}.
w_{Q1A}	Radial displacement of element "A" at juncture 1, due to Q_{1A}.
$\Theta_{PGASKET}$	Rotation of element "A" at juncture 1, due to P_{GASKET}.
$w_{PGASKET}$	Radial displacement of element "A" at juncture 1, due to P_{GASKET}.
Θ_{FS1A}	Rotation of element "A" at juncture 1, due to F_{S1A}.
w_{FS1A}	Radial displacement of element "A" at juncture 1, due to F_{S1A}.
Θ_{M1B}	Rotation of element "B" at juncture 1, due to M_{1B}.
w_{M1B}	Radial displacement of element "B" at juncture 1, due to M_{1B}.
Θ_{Q1B}	Rotation of element "B" at juncture 1, due to Q_{1B}.
w_{Q1B}	Radial displacement of element "B" at juncture 1, due to Q_{1B}.
Θ_{FS1B}	Rotation of element "B" at juncture 1, due to F_{S1B}.
w_{FS1B}	Radial displacement of element "B" at juncture 1, due to F_{S1B}.
Θ_{PGAS1B}	Rotation of element "B" at juncture 1, due to P_{GAS}.

w_{PGAS1B} .... Radial displacement of element "B" at juncture 1, due to P_{GAS}.
Θ_{M2B} .... Rotation of element "B" at juncture 2, due to M_{2B}.
w_{M2B} .... Radial displacement of element "B" at juncture 2, due to M_{2B}.
Θ_{Q2B} .... Rotation of element "B" at juncture 2, due to Q_{2B}.
w_{Q2B} .... Radial displacement of element "B" at juncture 2, due to Q_{2B}.
Θ_{FS2B} .... Rotation of element "B" at juncture 2, due to F_{S2B}.
w_{FS2B} .... Radial displacement of element "B" at juncture 2, due to F_{S2B}.
Θ_{PGAS2B} .... Rotation of element "B" at juncture 2, due to P_{GAS}.
w_{PGAS2B} .... Radial displacement of element "B" at juncture 2, due to P_{GAS}.
Θ_{M2C} .... Rotation of element "C" at juncture 2, due to M_{2C}.
w_{M2C} .... Radial displacement of element "C" at juncture 2, due to M_{2C}.
Θ_{Q2C} .... Rotation of element "C" at juncture 2, due to Q_{2C}.
w_{Q2C} .... Radial displacement of element "C" at juncture 2, due to Q_{2C}.
Θ_{FS2C} .... Rotation of element "C" at juncture 2, due to F_{S2C}.
w_{FS2C} .... Radial displacement of element "C" at juncture 2, due to F_{S2C}.
Θ_{PGAS2C} .... Rotation of element "C" at juncture 2, due to P_{GAS}.
w_{PGAS2C} .... Radial displacement of element "C" at juncture 2, due to P_{GAS}.
K_i i=1,32.......... Element stiffness from [2]

INTRODUCTION

The function of a reciprocating compressor cylinder head is to contain pressure that builds up inside the cylinder. The head is bolted to the compressor cylinder and is sealed with a gasket. An isometric cutaway view of a typical outer head is shown in figure 1.

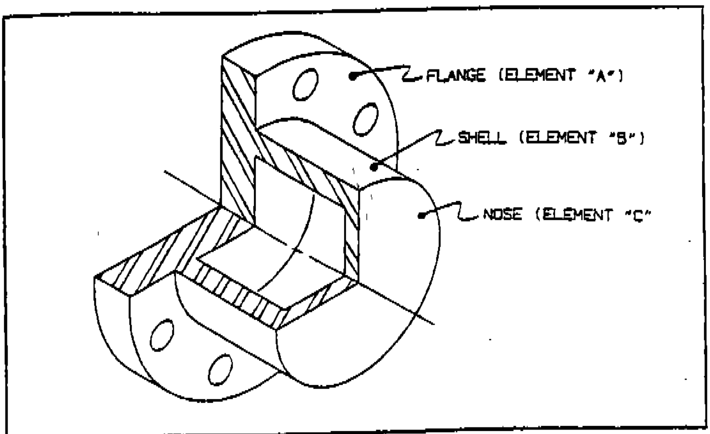

Figure 1. Isometric cutaway of typical outer head

Loading on this cylinder head consists of gas loading and bolting required to seal the gasket. Since the gas load is cyclic, two sets of stress calculations (at suction and discharge pressure) must be done to determine the fatigue factor of safety. This paper will address how to calculate stresses for a given pressure and gasket load.

Calculation of these stresses may appear simple given FEA capability. FEA is a very systematic and powerful tool for obtaining stresses; however, FEA is geometry driven and requires a "reasonableness" check on results. Thus, a systematic method, independent of FEA, using classical plate and shell theory [2] is used to create an initial geometry and check a given FEA result.

THEORY

This method is adapted from a method contained in ASME Code, Section 8, Division 2, Article 4-7, titled "Discontinuity Stresses" [1], using deformation equations from "Roark's Formulas for Stress and Strain" [2]. The method described in [1] is for a pressure vessel with internal pressure. This paper generalizes this method for any type of axisymmetric component with force and pressure loading.

The assumptions of this method are as follows:

1) The bolt load is treated as an annular load at a diameter equal to the bolt circle diameter. The bolts actually act as a number of individual loads evenly spaced around the circumference of the bolt circle.

2) Because of the assumption in 1) the head can be considered axisymmetric and can be analyzed as such.

3) The formulas used to calculate component shell stiffnesses [2] do not address thick-walled shells completely. Thus, with many practical applications of this method a thin-walled shell assumption must be made.

The analysis is initiated by separating the component into plate and shell elements for which deformations can be obtained. These separation points are referred to as junctures. The external pressure and force loads acting on the component are applied to these elements. Deformations at the element edges can be broken into 1) Radial displacements and 2) rotation of the shell or plate edge [1]. Internal moments and shear forces must exist on the element edges in order to maintain compatibility of deformations and restore continuity to the component [1]. At each juncture, these deformations are equated to solve for the internal moments and shear forces. In general, for n junctures, there are 2n unknown internal moments and shear forces, and 2n deformation equations. Equating the deformations to solve for the unknown moments and shears reduces to finding the appropriate stiffnesses and then finding the inverse of a 2n x 2n matrix containing these stiffnesses. This procedure is best demonstrated for the cylinder outer head in figure 2. This shows how the typical outer head shown in figure 1 would be analyzed. As indicated in figure 2, this head is acted on by gas pressure and gasket pressure. The total bolt force is equal and opposite to the force from the gasket and gas pressures. To facilitate the use of relations from [2] it is assumed that the bolt is a simple support. R_{BOLT} represents the force from this simple support. This assumption is also made when analyzing this head with the FEM.

At each juncture, two equations are written to express the equality of the combined deformations due to the external loads and the internal forces and moments. One equation will express the equality of the rotations, and the other will express the equality of displacements of adjacent elements.

For figure 2, the equations of the deformations at each element edge can be written:

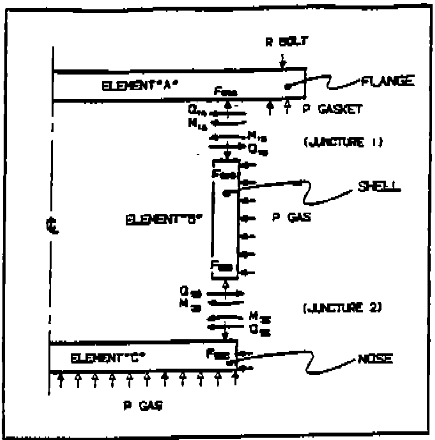

Figure 2. Separation of component into elements

Rotation of Element "A" at juncture 1, due to M_{1A}

$$\theta_{M1A}=\frac{1}{K_1}M_{1A} \qquad \text{Eqn. (1)}$$

Radial displacement of Element "A" at juncture 1, due to M_{1A}

$$\omega_{M1A}=\frac{1}{K_2}M_{1A} \qquad \text{Eqn. (2)}$$

Rotation of Element "A" at juncture 1, due to Q_{1A}

$$\theta_{Q1A}=\frac{1}{K_3}Q_{1A} \qquad \text{Eqn. (3)}$$

Radial displacement of Element "A" at juncture 1, due to Q_{1A}

$$\omega_{Q1A}=\frac{1}{K_4}Q_{1A} \qquad \text{Eqn. (4)}$$

Rotation of Element "A" at juncture 1, due to P_{GASKET}

$$\theta_{PGASKET}=\frac{1}{K_5}P_{GASKET} \qquad \text{Eqn. (5)}$$

Radial displacement of Element "A" at juncture 1, due to P_{GASKET}

$$\omega_{PGASKET}=\frac{1}{K_6}P_{GASKET} \qquad \text{Eqn. (6)}$$

Rotation of Element "A" at juncture 1, due to F_{S1A}

$$\theta_{FS1A}=\frac{1}{K_7}F_{S1A} \qquad \text{Eqn. (7)}$$

Radial displacement of Element "A" at juncture 1, due to F_{S1A}

$$\omega_{FS1A}=\frac{1}{K_8}F_{S1A} \qquad \text{Eqn. (8)}$$

Rotation of Element "B" at juncture 1, due to M_{1B}

$$\theta_{M1B}=\frac{1}{K_9}M_{1B} \qquad \text{Eqn. (9)}$$

Radial displacement of Element "B" at juncture 1, due to M_{1B}

$$\omega_{M1B}=\frac{1}{K_{10}}M_{1B} \qquad \text{Eqn. (10)}$$

Rotation of Element "B" at juncture 1, due to Q_{1B}

$$\theta_{Q1B}=\frac{1}{K_{11}}Q_{1B} \qquad \text{Eqn. (11)}$$

Radial displacement of element "B" at juncture 1, due to Q_{1B}

$$\omega_{Q1B}=\frac{1}{K_{12}}Q_{1B} \qquad \text{Eqn. (12)}$$

Rotation of Element "B" at juncture 1, due to F_{S1B}

$$\theta_{FS1B}=\frac{1}{K_{13}}F_{S1B} \qquad \text{Eqn. (13)}$$

Radial displacement of Element "B" at juncture 1, due to F_{S1B}

$$\omega_{FS1B}=\frac{1}{K_{14}}F_{S1B} \qquad \text{Eqn. (14)}$$

Rotation of Element "B" at juncture 1, due to P_{GAS}

$$\theta_{PGAS1B}=\frac{1}{K_{15}}P_{GAS} \qquad \text{Eqn. (15)}$$

Radial displacement of Element "B" at juncture 1, due to P_{GAS}

$$\omega_{PGAS1B}=\frac{1}{K_{16}}P_{GAS} \qquad \text{Eqn. (16)}$$

Rotation of Element "B" at juncture 2, due to M_{2B}

$$\theta_{M2B}=\frac{1}{K_{17}}M_{2B} \qquad \text{Eqn. (17)}$$

Radial displacement of Element "B" at juncture 2, due to M_{2B}

$$\omega_{M2B}=\frac{1}{K_{18}}M_{2B} \qquad \text{Eqn. (18)}$$

Rotation of Element "B" at juncture 2, due to Q_{2B}

$$\theta_{Q2B}=\frac{1}{K_{19}}Q_{2B} \qquad \text{Eqn. (19)}$$

Radial displacement of Element "B" at juncture 2, due to Q_{2B}

$$\omega_{Q2B}=\frac{1}{K_{20}}Q_{2B} \qquad \text{Eqn. (20)}$$

Rotation of Element "B" at juncture 2, due to F_{S2B}

$$\theta_{FS2B}=\frac{1}{K_{21}}F_{S2B} \qquad \text{Eqn. (21)}$$

Radial displacement of Element "B" at juncture 2, due to F_{S2B}

$$\omega_{FS2B}=\frac{1}{K_{22}}F_{S2B} \qquad \text{Eqn. (22)}$$

Rotation of element "B" at juncture 2, due to P_{GAS}

$$\theta_{PGAS2B}=\frac{1}{K_{23}}P_{GAS} \qquad \text{Eqn. (23)}$$

Radial displacement of Element "B" at juncture 2, due to P_{GAS}

$$\omega_{PGAS2B}=\frac{1}{K_{24}}P_{GAS} \qquad \text{Eqn. (24)}$$

Rotation of element "C" at juncture 2, due to M_{2C}

$$\theta_{M2C}=\frac{1}{K_{25}}M_{2C} \qquad \text{Eqn. (25)}$$

Radial displacement of Element "C" at juncture 2, due to M_{2C}

$$\omega_{M2C}=\frac{1}{K_{26}}M_{2C} \qquad \text{Eqn. (26)}$$

Rotation of element "C" at juncture 2, due to Q_{2C}

$$\theta_{Q2C}=\frac{1}{K_{27}}Q_{2C}$$ Eqn. (27)

Radial displacement of Element "C" at juncture 2, due to Q_{2C}

$$\omega_{Q2C}=\frac{1}{K_{28}}Q_{2C}$$ Eqn. (28)

Rotation of element "C" at juncture 2, due to F_{S2C}

$$\theta_{FS2C}=\frac{1}{K_{29}}F_{S2C}$$ Eqn. (29)

Radial displacement of Element "C" at juncture 2, due to F_{S2C}

$$\omega_{FS2C}=\frac{1}{K_{30}}F_{S2C}$$ Eqn. (30)

Rotation of element "C" at juncture 2, due to P_{GAS}

$$\theta_{PGAS2C}=\frac{1}{K_{31}}P_{Gas}$$ Eqn. (31)

Radial displacement of Element "C" at juncture 2, due to P_{GAS}

$$\omega_{PGAS2C}=\frac{1}{K_{32}}P_{Gas}$$ Eqn. (32)

K_i are the element stiffnesses and are a function of element geometry, element material, loading, and boundary conditions. Both external loads (P_{GASKET} and P_{GAS}) are known. When finding deformations from [2], R_{BOLT} is assumed to be the reaction force from a simple support. All internal forces and moments are unknown, namely, F_{S1A}, F_{S1B}, F_{S2B}, F_{S2C}, M_{1A}, Q_{1A}, M_{1B}, Q_{1B}, M_{2B}, Q_{2B}, M_{2C}, and Q_{2C}. Thus there are 12 unknowns.

To maintain equilibrium the following must be true:

$$F_S=F_{S1A}=F_{S1B}=F_{S2B}=F_{S2C}$$ Eqn. (33)

F_S can be determined by a summation of the forces in the vertical direction on element "C". Also, from continuity the following must be true:

$$M_1=M_{1A}=M_{1B}$$ Eqn. (34)

$$Q_1=Q_{1A}=Q_{1B}$$ Eqn. (35)

$$M_2=M_{2B}=M_{2C}$$ Eqn. (36)

$$Q_2 = Q_{2B} = Q_{2C} \qquad \text{Eqn. (37)}$$

This reduces the number of unknowns from 12 to 4, leaving the unknowns M_1, Q_1, M_2, and Q_2. These four unknowns are solved by equating the deformation and rotations from each element at each juncture. This will give two equations at each of two junctures for a total of four equations. These four equations are used to solve for the four unknowns.

At juncture 1, equating the rotations of elements "A" and "B":

$$\theta_{M1A}+\theta_{Q1A}+\theta_{PGASKET}+\theta_{FS1A}=\theta_{M1B}+\theta_{Q1B}+\theta_{FS1B}+\theta_{PGAS1B} \qquad \text{Eqn. (38)}$$

At juncture 1, equating the radial displacements of elements "A" and "B":

$$\omega_{M1A}+\omega_{Q1A}+\omega_{PGASKET}+\omega_{FS1A}=\omega_{M1B}+\omega_{Q1B}+\omega_{FS1B}+\omega_{PGAS1B} \qquad \text{Eqn. (39)}$$

At juncture 2, equating the rotations of elements "B" and "C":

$$\theta_{M2B}+\theta_{Q2B}+\theta_{FS2B}+\theta_{PGAS2B}=\theta_{M2C}+\theta_{Q2C}+\theta_{FS2C}+\theta_{PGAS2C} \qquad \text{Eqn. (40)}$$

At juncture 2, equating the radial displacements of elements "B" and "C":

$$\omega_{M2B}+\omega_{Q2B}+\omega_{FS2B}+\omega_{PGAS2B}=\omega_{M2C}+\omega_{Q2C}+\omega_{FS2C}+\omega_{PGAS2C} \qquad \text{Eqn. (41)}$$

Substituting Equations 1 through 37 into Equations 38,39,40 and 41 and simplifying:

At juncture 1, equating the rotations of elements "A" and "B":

$$\frac{M_1}{K_1}+\frac{Q_1}{K_3}+\frac{P_{GASKET}}{K_5}+\frac{F_S}{K_7}=\frac{M_1}{K_9}+\frac{Q_1}{K_{11}}+\frac{F_S}{K_{13}}+\frac{P_{GAS}}{K_{15}} \qquad \text{Eqn. (42)}$$

At juncture 1, equating the radial displacements of elements "A" and "B":

$$\frac{M_1}{K_2}+\frac{Q_1}{K_4}+\frac{P_{GASKET}}{K_6}+\frac{F_S}{K_8}=\frac{M_1}{K_{10}}+\frac{Q_1}{K_{12}}+\frac{F_S}{K_{14}}+\frac{P_{GAS}}{K_{16}} \qquad \text{Eqn. (43)}$$

At juncture 2, equating the rotations of elements "B" and "C":

$$\frac{M_2}{K_{17}}+\frac{Q_2}{K_{19}}+\frac{F_S}{K_{21}}+\frac{P_{GAS}}{K_{23}}=\frac{M_2}{K_{25}}+\frac{Q_2}{K_{27}}+\frac{F_S}{K_{29}}+\frac{P_{GAS}}{K_{31}} \qquad \text{Eqn. (44)}$$

At juncture 2, equating the radial displacements of elements "B" and "C":

$$\frac{M_2}{K_{18}}+\frac{Q_2}{K_{20}}+\frac{F_S}{K_{22}}+\frac{P_{GAS}}{K_{24}}=\frac{M_2}{K_{26}}+\frac{Q_2}{K_{28}-K_{30}}+\frac{F_S}{}+\frac{P_{GAS}}{K_{32}} \qquad \text{Eqn. (45)}$$

Simplifying:

$$M_1\frac{(K_9-K_1)}{(K_1K_9)}+Q_1\frac{(K_{11}-K_3)}{(K_3K_{11})}=F_S\frac{(K_7-K_{13})}{(K_{13}K_7)}-\frac{P_{GASKET}}{K_5}+\frac{P_{GAS}}{K_{15}} \qquad \text{Eqn. (46)}$$

$$M_1\frac{(K_{10}-K_2)}{(K_2K_{10})}+Q_1\frac{(K_{12}-K_4)}{(K_4K_{12})}=F_S\frac{(K_8-K_{14})}{(K_{14}K_8)}-\frac{P_{GASKET}}{K_6}+\frac{P_{GAS}}{K_{16}} \qquad \text{Eqn. (47)}$$

$$M_2\frac{(K_{25}-K_{17})}{(K_{17}K_{25})}+Q_2\frac{(K_{27}-K_{19})}{(K_{19}K_{27})}=F_S\frac{(K_{21}-K_{29})}{(K_{29}K_{21})}+P_{GAS}\frac{(K_{23}-K_{31})}{(K_{31}K_{23})} \qquad \text{Eqn. (48)}$$

$$M_2\frac{(K_{26}-K_{18})}{(K_{18}K_{26})}+Q_2\frac{(K_{28}-K_{20})}{(K_{20}K_{28})}=F_S\frac{(K_{22}-K_{30})}{(K_{30}K_{22})}+P_{GAS}\frac{(K_{24}-K_{32})}{(K_{32}K_{24})} \qquad \text{Eqn. (49)}$$

If we define the following constants:

$$C_{I,J}=\frac{K_J-K_I}{K_IK_J};\; C_k=\frac{1}{K_k} \qquad \text{Eqn. (50)}$$

Simplifying again:

$$M_1C_{1,9}+Q_1C_{3,11}=F_sC_{13,7}-P_{GASKET}C_5+P_{GAS}C_{15} \qquad \text{Eqn. (51)}$$

$$M_1C_{2,10}+Q_1C_{4,12}=F_SC_{14,8}-P_{GASKET}C_6+P_{GAS}C_{16} \qquad \text{Eqn. (52)}$$

$$M_2C_{17,25}+Q_2C_{19,27}=F_SC_{29,21}+P_{GAS}C_{31,23} \qquad \text{Eqn. (53)}$$

$$M_2C_{18,26}+Q_2C_{20,28}=F_SC_{30,22}+P_{GAS}C_{32,24} \qquad \text{Eqn. (54)}$$

Putting this in matrix form:

$$\begin{bmatrix} C_{1,9} & C_{3,11} & 0 & 0 \\ C_{2,10} & C_{4,12} & 0 & 0 \\ 0 & 0 & C_{17,25} & C_{19,27} \\ 0 & 0 & C_{18,26} & C_{20,28} \end{bmatrix} \begin{Bmatrix} M_1 \\ Q_1 \\ M_2 \\ Q_2 \end{Bmatrix} = \begin{Bmatrix} C_{13,7}F_S-C_5P_{GASKET}+C_{15}P_{GAS} \\ C_{14,8}F_S-C_6P_{GASKET}+C_{16}P_{GAS} \\ C_{29,21}F_s+C_{31,23}P_{GAS} \\ C_{30,22}F_s-C_{32,24}P_{GAS} \end{Bmatrix} \qquad \text{Eqn. (55)}$$

or

$$[C]\{M\}=\{B\} \qquad \text{Eqn. (56)}$$

The zeros in the matrix are due to the fact that deformation of element B at juncture 2 due to the loads on element B at juncture 1 are ignored. This is also true of the deformation of element B at juncture 1 due to the loads on element B at juncture 2. They are ignored because they are an order of magnitude less than the other deformations. Experience with this method has shown this assumption does not change the results appreciably.

To solve for {M} we multiply both sides of this equation by the inverse of [C].

$$[C][C]^{-1}\{M\}=\{B\}[C]^{-1} \qquad \text{Eqn. (57)}$$

$$\{M\}=\{B\}[C]^{-1} \qquad \text{Eqn. (58)}$$

With {M} now determined, M_1, Q_1, M_2, Q_2, are known. Superposition is used to calculate the stresses due to each of these internal and external loads. Stresses on each of these simple elements can be found in reference [2].

DISCUSSION

The advantage of this method is that it is very systematic for a given type of geometry. This procedure has been incorporated into a computer program where an initial estimate of section thicknesses can be made. After these initial sections are determined, appropriate factors of safety are calculated. These section thicknesses can be adjusted depending on the factor of safety. This procedure will iterate until the desired factor of safety has been reached. The advantage of iterating with this procedure over FEA is that this procedure is much less CPU intensive than FEA procedures. When the desired factor of safety has been reached with this method, the associated geometry can now be used to start an FEA run. Typically only one run will have to be made using FEA. Finally, when the FEA is completed the "reasonableness" of the analysis has already been verified by this independent analysis.

A disadvantage of this method is that the stiffness (K_i) calculations [2] are very tedious and time consuming. Also, great care must be taken in determining the sign convention to be used in determining the deformations from [2]. Another limitation of this method is that all deformations and stresses calculated for shells [2] are based on thin-walled shell formulas. Often this is not the case in the design of these components. But, experience has shown that use of these thin shell formulas appears to be conservative. Further research must be done to determine deformations in thick-walled shells for use with this method.

CONCLUSION

This paper generalizes an ASME Code procedure for pressure vessels with internal pressure for parts with both pressure and force loads. This procedure enables one to simplify a complex structure with plate and shell elements and calculate the stresses in each element independently. The stiffness and stresses from this generalized method can be calculated from [2].

There are many advantages to this method. This method is systematic and serves as an independent check for FEA. It will reduce the number of iterations that must be done with FEA to reach a final design. It also can serve as design tool for designers that may not have access to FEA software.

A disadvantage of this method is that it is tedious and great care must be taken when specifying sign conventions. Another disadvantage is that most of the deformations for shells from [2] are for thin walls only. Caution should be used when using this for thick-walled shells.

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REFERENCES

1) ASME Pressure Vessel Code, 1986 Edition, Section VIII, Division 2, Article 4-7, " Discontinuity Stresses".

2) Young, Warren C.,"Roark's Formulas for Stress and Strain", McGraw-Hill, 1989.