Pressure vessel design using boundary element method with optimization

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Abstract

Design of pressure vessels is covered by references such as the ASME Pressure Vessel Code and textbooks devoted to pressure vessel design. Detailed stress analysis, particularly in the area of discontinuities, is generally left to the design engineer. The type discontinuity addressed in this paper is the design of bolted flanges for a pressure vessel. Work for this project involves optimization of the hub contour using the ASME Pressure Vessel Code requirements as constraints. This paper summa&es the initial work, development of the BEM (BEASY) models and verification of the model to classical techniques such as a "Roark" [123. This two-dimensional model will later be expanded to a full, three-dimension model; and, will also provide data for establishing allowable defect sizes, based upon inspection techniques and design life.

Pressure vessel design

Design of pressure vessels is governed by the ASME pressure vessel code [1]. Other textbooks such as Farr [2], Moss [3], Chuse [4], Harvey [5], Bednar [6], and Gill [7], provide valuable insight and guidelines to pressure vessel design. Bolted flange analysis is discussed in machine design textbooks such- as Norton [8] or speciality books such as Bickford or Blake [9, 10] or Company design practices or criteria. Detailed stress analysis, particularly in the area of discontinuities, is generally left to the design engineer. Different type stresses are defined by Appendix 4 of the ASME code [1]. Stress limits, (allowable stress magnitudes), based upon the type stress, are addressed by Appendix 4 (*Mandatory Design Based on Stress Analysis*) of the ASME code [1]. These are discussed later in this paper.

The type discontinuity addressed in this paper is one associated with design of bolted flanges for a pressure vessel. Figure la illustrates one type design which consists of the shell welded to the flange. Figure lb illustrates another type design, a hubbed flange, with the weld located away from the shell/flange discontinuity. This is done to locate the weld in an area of lower bending stress, improving the strength of the joint; and, also to locate it in an area where it may require less weld material (cost) and can be more easily inspected.

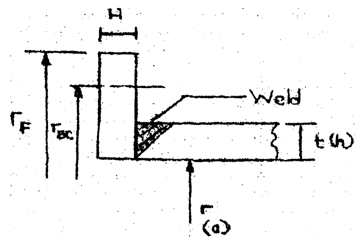

Figure la - Basic Configuration

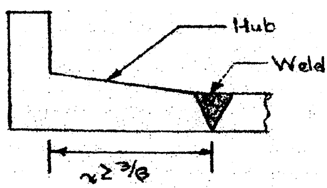

Figure lb - **Hubbed Configuration**

r_F	Flange Outer Radius
r_{BC}	Bolt Circle Radius
r	Shell (Vessel) Inner Radius
t or h	Shell (Wall) Thickness
H	Flange Thickness
x	Distance From Joint (Discontinuity) to Weld

Bending moments at a discontinuity, such as a flange, will generally be local and diminish in magnitude as the distance from the discontinuity is increased. Referring to Timoshenko[11], when the term βx = 3.0, the moment effect is almost zero.

$$\beta^4 = \frac{3(1-\upsilon^2)}{(ah)^2}$$

$$\beta x = 3.0$$

v	Poisson's Ratio
a	Shell Radius
h	Vessel (Shell) Thickness
x	Axial Distance from Discontinuity

Using the ASME Code, previously mentioned references, and handbooks such as Roark[12] pressure vessel design could be a very complex task. With the advent of the computer age, techniques such as the finite element method (FEM) and boundary element method (BEM) became very valuable design and analysis aids.

Section VIII, Division 2 of the ASME Code defines several category of stresses: Primary, Secondary, and Peak. A *primary stress* is a normal or shear stress developed by the imposed loading and necessary to satisfy the laws of equilibrium, such as the hoop (primary membrane) stress resulting from internal pressure in a shell. *Secondary stresses* are normal or shear stress developed by the constraint of adjacent parts or the self-constrain of a structure, a bending stress at a gross structural discontinuity. The basic characteristic of a secondary stress is it is self-limiting, local yielding and minor distortions can satisfy the conditions which cause to stress to occur. A *peak stress* is a stress which does not cause any noticeable distortion and is undesirable because it may be a possible source of a fatigue crack or brittle fracture, for example, the stress at a local structural discontinuity. Definitions of all terms and tables of combinations and allowable stresses are provided in Appendix 4 of Section VIII, Division 2. Provisions for plastic, limit, experimental, shakedown, and fatigue analysis are also available in this same section of the Code.

Coupled with the stress analysis are design considerations for failure analysis. Allowable defect limits must be determined by the designer, even with the Code allowable limits. Teak before failure", design life, damage tolerance are all factors which must be considered in the design process. Once again textbooks and handbooks, such as Collins [13], Dowling [14], Ta& and Paris [15], Maddox [16], Brooks and Choudhury [17], and Barson and Rolfe [18] are available for design reference.

Boundary element analysis with optimization

The use of BEM is not new to pressure vessel analysis as evidenced by

Trevelyan [19] and Floyd [20]. Fracture and crack growth using BEM is also evidenced by textbooks such as Prasad [2 1], Aliabadi [22][23], Monahan [24], and Leitao [25]. A test model, as shown by Figure 2, has been run to verify BEM results. The model was based upon a flanged and bolted pipe. The model was verified using equations from Table XIII, case 32 of Roark [12]. Results of this model were comparable to the "hand calculations" of Roark

Like the boundary element method, optimization techniques have been enhanced by the continued growth of computers. Vanderplaats [26] is one of those who has been part of this growth as well as others such as Chanrupatala and Belegundu [27]. The work being performed in conjunction with this paper is based upon design of a hubbed flange, subject to the requirements of the ASME Code, using the boundary element method. BEASY® [28] and VisualDOC® [29] are the software codes selected for this work In other words, what is being accomplished is to optimize a design based upon BEM analysis with constraints imposed by codes such as the ASME Pressure Vessel Code.

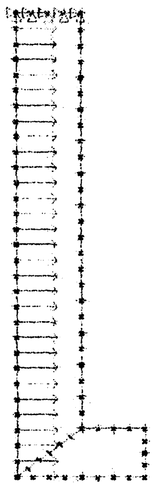

Figure 2 -
BEASY Model

A criterion or objective "function" must be determined which will satisfy inequality; and, possibly, equality constraints. In turn, the objective must be minimized (or maximized, depending upon the problem.) For a simple function, this means determining where the first derivative is zero, and if the second derivative is positive or negative at those points. Furthermore, at those points, the design or **objective must satisfy** all constraints. **That is,** the **solution** must ***be feasible.***

The actual problem is somewhat more complex. The first question becomes what is to be minimized? In this case, it will be weight. Although, with sufficient time and thought, this can be translated into cost, considering fabrication costs, inspection costs, and material costs. Weight should provide a good working model. Constraints will be to satisfy the stress limits of the ASME Code and the weld to be in a low bending stress area ($\beta x \geq 3.0$). As the work progresses, cost and multi-objective function problems will be developed. The final step will be to introduce crack propagation constraints into the models.

Conclusions

Initial results show theboundary element method will provide accurate predictions of the stresses in a pressure vessel flange. With further development, an optimum hub contour and weld location, subject to pressure vessel code and other constraints will be obtained. Once this methodology has been established, the work will be expanded to three-dimensional models. Flange-opening under loads can then be considered, with local stiffening, as a function of angular location, a consideration in the methodology to be developed.

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