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**Elastic Structural Analysis of the
Pin-Type Solid Breeder Blanket First Wall**

George E. Orient and Naur M. Ghoniem

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1. INTRODUCTION

A key to the successful design of a fusion blanket is the distribution of the amount of structural material in the first wall and the blanket enclosing structure. The first wall must support dynamic and static mechanical loads and at the same time sustain the thermal gradients and erosion resulting from the plasma heat and particle fluxes. Dynamic and static loads require high flexural rigidity, which is influenced by flow channels designed to cool the high heat flux incident on the first wall. Since the first wall temperature is roughly proportional to its thickness, thermal stresses require a thin first wall. On the other hand, a thick wall is necessary to withstand surface erosion due to particle fluxes. A feasible design must therefore represent an optimal solution satisfying such conflicting requirements.

The structural design of the breeder blanket must not only satisfy the steady state operating conditions, but must also meet cyclic conditions due to start-ups, shut-downs and non-steady operation. Fatigue of the first wall, crack propagation under cyclic load and residual stress build-up are some of the most important issues related to repetitive loading conditions. Another problem arising from the time history of the loads is thermal shock due to power ramps. One also has to consider emergency situations such as loss of pressure.

Throughout its life, the blanket will be subjected to a radiation environment causing radiation induced creep, swelling, and embrittlement. Although creep has a tendency to relieve high stresses due to thermal gradients, it can cause large displacements. Under axially restrained conditions creep buckling and high stresses due to swelling may be important. Loss of the structural material due to erosion must also be considered so that the module can sustain the loads under end of life conditions. A satisfactory structural design requires that all of these operating conditions be met without loss of structural integrity of the system.

To address some of the issues described above and gain some insight into scaling, a displacement type finite element model, based on classical beam theory, has been developed.

2. MODELING

The first wall structure is modeled here as an assembly of curved I beams. Then the resulting thermoelasticity problem is solved using a displacement type finite element formulation. The curved beam is discretized in 2D-frame elements. The following assumptions have been adopted:

1. A homogeneous, isotropic, linearly elastic material,
2. Linear thermal expansion,
3. Cross sectional dimensions are small compared to the radius of curvature.

2.1 Stiffness Matrix and Consistent Load Vectors for a 2-D Frame Element

When deriving a displacement type finite element formulation, one has to choose a displacement field which satisfies the following conditions:

- it can describe the rigid body motions of a particular element,
- it has sufficient freedom to allow kinematic continuity between the elements,
- it can produce a displacement pattern corresponding to the constant strain state of the elements. This requirement is necessary for the element in order to be able to converge to the true strain state upon refining the partition of a body in a general strain state.

Consider a two-dimensional frame element given in Fig. 1. Its behavior for small displacements is a superposition of that of a bar and a beam. Let $u(x)$ and $v(x)$ denote the axial and transverse displacements in the local coordinate system. Later quantities will be expressed in a global coordinate system. The global quantities will be denoted by a prime. The nodal displacement quantities associated with a frame element are the axial and transverse components of the motion, u , v and the rotation at the nodal points, β . The corresponding force and moment quantities are F_x , F_y and M . Subscripts 1 and 2 refer to the arbitrarily chosen "beginning" and the "end" of the element. Transverse and axial distributed line loads are denoted by p and q , respectively.

Fig. 1. Forces and displacements on a 2-D frame element

Consider a linear interpolation for $u(x)$ and a cubic interpolation for $v(x)$. The rigid body motion of a bar is a translation which is clearly included in a linear displacement field. Since the nodal degrees of freedom are u_1 and u_2 in the axial direction, inter-element continuity is guaranteed. For a bar, the extensional strain is equal to the first derivative of the axial displacement field; consequently, a linear interpolation function permits a constant strain state. The rigid body motions of a beam are uniform transverse translation and a rotation which are included in a cubic field.

There are four generalized nodal degrees of freedom, because classical beam theory requires that the displacement and the slope to be continuous between elements. This makes it necessary to choose at least a cubic interpolation function. The constant strain mode for thin beams is a constant curvature, which requires a displacement at least quadratic in the axial coordinate. Thus,

$$u(x) = N_u^T u_n = [1-\xi] \begin{bmatrix} u_1 \\ u_2 \end{bmatrix} \quad (1)$$

$$v(x) = N_v^T u_n = [1-3\xi^2+2\xi^3 \quad d(\xi-2\xi^2+\xi^3) \quad 3\xi^2-2\xi^3 \quad d(-\xi^2+\xi^3)] \begin{bmatrix} v_1 \\ \beta_1 \\ v_2 \\ \beta_2 \end{bmatrix} \quad (2)$$

where ξ denotes x/d and d is the length of the element. The axial strain is expressed in terms of the displacement field as follows (Euler-Bernoulli beam theory):

$$\epsilon_x = \frac{du}{dx} - y \frac{d^2v}{dx^2} \quad (3)$$

The axial stress is given by linear stress-strain relations:

$$\sigma_x = E(\epsilon_x - \alpha T) \quad (4)$$

And the strain energy is:

$$U = \frac{1}{2} (\epsilon_x - \alpha T) \sigma_x = \frac{E}{2} (\epsilon_x - \alpha T)^2 \quad (5)$$

The total potential energy of the element is defined as:

$$V = U + V_E \quad (6)$$

where the potential energy of the external forces is given by:

$$V_E = -F_{x1}u_1 - F_{x2}u_2 - \int_0^d p(x)u(x)dx - F_{y1}v_1 - F_{y2}v_2 - M_1\beta_1 - M_2\beta_2 - \int_0^d q(x)v(x)dx \quad (7)$$

Applying the theorem of minimum potential energy, $\delta V = 0$ leads to the complete stiffness relation:

$$\begin{bmatrix} X_1 \\ Y_1 \\ M_1 \\ X_2 \\ Y_2 \\ M_2 \end{bmatrix} = \underline{K} \begin{bmatrix} u_1 \\ v_1 \\ \beta_1 \\ u_2 \\ v_2 \\ \beta_2 \end{bmatrix} = \underline{P} - \underline{P}_T \quad (8)$$

$$\underline{K} = E/d \begin{bmatrix} A & 0 & 0 & -A & 0 & 0 \\ & 12/d^2 & 6I/d & 0 & -12/d^2 & 6I/d \\ & & 4I & 0 & -6I/d & 2I \\ & \text{symmetric} & & A & 0 & 0 \\ & & & & 12I/d^2 & -6I/d \\ & & & & & 4I \end{bmatrix} \quad (9)$$

$$\underline{P} = \begin{bmatrix} pd/2 \\ qd/2 \\ qd^2/12 \\ -pd/2 \\ qd/2 \\ -pd^2/12 \end{bmatrix} \quad \underline{P}_T = \begin{bmatrix} -F_T \\ 0 \\ M_T \\ -F_T \\ 0 \\ -M_T \end{bmatrix} \quad (10)$$

$$F_T = \int_A \alpha T(y,z) \, dA$$

$$M_T = \int_A y \alpha T(y,z) \, dA$$

where A denotes the area of the cross section and I is the moment of inertia. The transverse and axial loads are taken constant over the length of an element to obtain the consistent load vector in (10). The temperature field $T(x,y,z)$ has been replaced by

$$T(y,z) = \frac{1}{d} \int_0^d T(x,y,z) \, dx \quad (11)$$

which does not affect convergence if one takes fine enough partitions to capture the change of the thermal field in the x direction. It can be observed that thermal effects appear as a consistent load vector. It can also be shown that the stiffness matrix, $\underline{K}$ exhibits all the necessary rigid body motions and constant strain modes as it can be anticipated from the orders of interpolations used to describe the displacement field.

For practical applications, it is convenient to express the nodal displacements and forces in terms of global coordinates. The transformation is given by the following relations:

$$\begin{bmatrix} u' \\ v' \\ \beta' \end{bmatrix} = \begin{bmatrix} \cos \theta & \sin \theta & 0 \\ -\sin \theta & \cos \theta & 0 \\ 0 & 0 & 1 \end{bmatrix} \begin{bmatrix} u \\ v \\ \beta \end{bmatrix} = \underline{T}_1 \begin{bmatrix} u \\ v \\ \beta \end{bmatrix} \quad (12)$$

$$\underline{u}' = \begin{bmatrix} u'_1 \\ v'_1 \\ \beta'_1 \\ u'_2 \\ v'_2 \\ \beta'_2 \end{bmatrix} = \begin{bmatrix} \underline{T}_1 & \\ & \underline{T}_1 \end{bmatrix} \begin{bmatrix} u_1 \\ v_1 \\ \beta_1 \\ u_2 \\ v_2 \\ \beta_2 \end{bmatrix} = \underline{T} \underline{u} \quad (13)$$

$$\underline{P}' = \underline{T} \underline{P}$$

$$\underline{P}'_T = \underline{T} \underline{P}_T$$

$$\underline{K}' = \underline{K}^T \underline{K} \underline{T} \quad (14)$$

Application of 2D frame elements to analyzing thin planar structures: boundary conditions.

In order to apply the previous method to the analysis of the first wall, we first define a discretization method of its structure. Then, it is assumed that the elements are interconnected at nodal points, and therefore, system stiffness relation is established. This takes a form similar to that of an element stiffness relation,

$$\underline{K} \underline{u} = \underline{F} \quad (15)$$

where $\underline{K}$ is a banded matrix. The bandwidth is determined by the numbering of the nodes.

Next, we modify the stiffness relation by including boundary conditions. For example, built-in conditions are achieved by setting both displacements and slope to zero at the boundary, i.e.,

$$u = 0, v = 0, \beta = 0 \quad (16)$$

while symmetry with respect to a horizontal axis is modeled by setting the axial displacement and the slope zero,

$$v = 0, \beta = 0 \quad (17)$$

Finally, a simple support condition is such that only rotation is allowed:

$$u = 0, v = 0 \quad (18)$$

To delete the i^{th} degree of freedom, we simply zero out the i^{th} row and i^{th} column of the system and replace K_{ii} by one. This guarantees that upon solving the equation, the i^{th} displacement component will be zero. Using element stiffness relations, we then solve for nodal forces and hence nodal stresses.

The modeling process described above is applied as follows: Taking advantage of symmetry one half of the first wall in one blanket module is considered. This is realistic, since the blanket structure and the loads are

symmetric. The corresponding boundary conditions are (16) at the location where the neighboring modules meet and (17) at the middle of the segment. To discretize the structure, we divided it into five segments. Calculations to check accuracy using ten segments are also performed. It is found that approximately five segments give an accurate representation of the stress distribution. Integrals appearing in the expressions for thermal forces and moments are calculated by Simpson's rule using 12 partitions of the cross sections.

3. RESULTS OF CALCULATIONS

Fig. 2 shows the pure pressure stresses at beginning of life. It can be observed that the stresses are positive due to the internal pressure and the support gives rise to local bending. As a result, the stresses at the plasma side tend to be less positive, while the inner side stresses are increased. The design limit for the primary stresses at the operating temperature is approximately 170 MPa. To allow for a sacrificial layer for erosion, the plasma side of the first wall is thickened.

Results for thermal and pressure stresses for this design, based upon a temperature profile without a central fin as discussed in reference [1], are given on Fig. 3.-6. Based on the assumption that the support behind the manifold allows thermal expansion of the whole unit, the base temperature for the thermal strain is taken as the inlet coolant temperature. The plasma side of the first wall experiences compression due to the higher temperatures. The relatively high level of the stresses results from the interaction between the high bulk temperature and the supports, as well as the gradients across the first wall. It is found that bulk temperature is the more important factor here, since azimuthal variation of the stresses was not large.

In order to reduce the bulk temperature, fins in the coolant channels and surface roughening have been employed to improve heat transfer between the coolant and the structure. Fig. 3. and 4. show total stresses with one fin in the coolant channel and with and without surface roughening. Improvements due to surface roughening are relatively because only a small fraction of the total surface is assumed to be roughened. On Fig. 5. and 6. the effect of surface roughening can be observed when there are two fins present in the coolant channel. The stresses are lower than the values for the single fin configuration, primarily because of the lower bulk temperature. The values are relatively high, but referring to the design guidelines for secondary stresses published in reference [3] they are allowable. At operating temperature the allowable primary stress for C-9 is about 170 MPa. The design limit for secondary stresses which include thermal stresses is three times the value for the primary stresses.

Fig. 7 illustrates what the pressure stresses might look like at the end of life assuming three full-power years of operation and 1 mm/year erosion rate. The point here is that the pressure stresses are below the allowable limit. Stresses shown here are to be taken only as estimates, since many processes neglected here may significantly alter the elastic state of the structure.

plasma side

PRESSURE STRESSES IN THE FIRST WALL
BEGINNING OF LIFE

Fig. 2.

PRESSURE AND THERMAL STRESSES IN THE FIRST WALL
BEGINNING OF LIFE
ONE FIN, NO ROUGHNESS

Fig. 3.

PRESSURE AND THERMAL STRESSES IN THE FIRST WALL
BEGINNING OF LIFE
ONE FIN WITH ROUGHNESS

Fig. 4.

PRESSURE AND THERMAL STRESSES IN THE FIRST WALL
BEGINNING OF LIFE
TWO FINS, NO ROUGHNESS

Fig. 5.

**PRESSURE AND THERMAL STRESSES IN THE FIRST WALL
BEGINNING OF LIFE
TWO FINS WITH ROUGHNESS**

Fig. 6.

PRESSURE STRESSES IN THE FIRST WALL
END OF LIFE

Fig. 7.

Fig. 8. Side-wall modeling process

4. ANALYSIS OF THE SIDE-WALL

In order to attain inherent structural safety, the design of the blanket must be failure tolerant. Loss of pressure or rupture of a module must not start a catastrophic chain of failures propagating through the whole blanket. As the worst case one can consider a single module loaded by the internal pressure of the coolant, while the neighboring module suffered total loss of pressure. In order to be safe, a module has to be able to support the pressure without the equalizing pressure on the side walls resulting from the presence of the adjacent pressurized module. In principle, a tapered side wall could result in an optimal load carrying structure and minimal impact on the blanket performance, since the added structural material would appear at the back of the blanket. However, the loads and the geometry of our design would require excessively thick side wall at the inlet manifold. Alternatively, periodic supports as shown on Fig. 8 along the axis of the module can add the necessary stiffness to the side wall. The necessary spacing can be estimated as follows:

Consider the rectangular segment of the side wall bounded by the manifold, the supports and the cylindrical part of the first wall. Relaxing the connectivity conditions at the top of the segment, the boundary conditions are those shown in Fig. 8. (b). Further relaxing the symmetry conditions at the vertical edges of the segment, simple supported conditions can be used. This modeling process is rather conservative, but will yield a safe geometry. As it turns out, it is necessary to assume a mechanical coupling between the side walls of the adjacent modules in order to achieve a flexural rigidity that permits reasonable spacing between the supports.

Fig. 9. Cross section geometry of the coupled side-wall

The flexural rigidity of a double-channelled cross section as shown on Fig. 9. is given by

$$D = \frac{E}{1-\nu^2} \int_A y^2 dy = \frac{E}{1-\nu^2} ((h_1 + h_2 + h_3)^3 - (h_2 + h_3)^3 + h_3^3) \quad (19)$$

after neglecting the walls of the channels.

The maximum stress occurs in the middle of the free edge and at the outer fiber it is given by [2]

$$\sigma = \frac{E(h_1 + h_2 + h_3)}{2(1-\nu^2)} \frac{b^2}{D(b/a)^2} p\beta \quad (20)$$

where p is the coolant pressure, a is the distance between supports, b/a is the aspect ratio of the rectangular plate and β is a numerical factor tabulated for different aspect ratios on page 210 of reference [2]. For the current design $h_1=0.025$ m, $h_2=0.01$ m and $h_3=0.005$ m and $b=0.6$ m resulting $D=7.84$ MPa m². An aspect ratio $b/a=1/3$ yields 251 MPa maximum stress. Noting that it occurs at the middle of the free end in the model, which is a connection line to the cylindrical surface in the real structure this value is acceptable. Safer still, if $b/a=1/2$ then the maximum stress is 110 MPa, therefore supports at every 1.2 m will provide enough rigidity.

5. DISCUSSION OF LIMITATIONS AND UNCERTAINTIES

While this model is based upon a finite element model to the first wall geometry, it has not accounted for several important aspects. Future efforts may be necessary to resolve the importance of these aspects, which are discussed below:

1. Further elastic-plastic analysis is necessary to better understand the scaling of geometry, since the stresses exceed the elastic limit. Furthermore, residual stress build-up due to cyclic load should also be calculated.
2. The built-in end condition at the joints with the side walls is valid, since there is no rotation and translation in the radial direction due to symmetry. Translation in the tangential direction is negligible compared to displacements in other locations, because temperatures are the lowest at the joint. Application of symmetry conditions in a continuous curved beam at the top of the arch may not be realistic, depending upon the real geometry at this point. If a channel is present there, a hinge type boundary condition may be a better approximation, if a significant portion of the cross section is removed. In this case, a detailed analysis is necessary to quantify stress concentrations around coolant channels.
3. Swelling is a significant phenomenon for an axially restrained structure, such as a curved member. The equivalent thermal field arising from swelling is superposed on the real temperatures and can cause high stresses.
4. Since the structure is made of a ferritic alloy (9-C), significant body

forces may be present due to the magnetic field, and proper analysis should address this question. If the distribution of the magnetic body force field is known then the tangential and normal components can be calculated and input as distributed transverse and axial loads in equation (10).

5. Since the temperatures throughout the first wall structure are high, both thermal and irradiation creep are expected to play significant role during the lifetime of the blanket. A treatment of the creep contribution is discussed in reference [4] and is directly applicable to this problem.
6. Swelling strains could be easily incorporated similarly to the consistent loads due to thermal strains given in equation (10).
7. Time dependent erosion can be included when a total time analysis is performed by changing the cross section geometry as the solution continues.

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