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by

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# THE RESPONSE OF A THERMAL BARRIER SYSTEM TO ACOUSTIC EXCITATION IN A GAS TURBINE NUCLEAR REACTOR

W. S. Betts, Jr.
General Atomic Company
San Diego, California

R. D. Blevins
General Atomic Company
San Diego, California
Member ASME

## ABSTRACT

A gas turbine (GT) located within a High-Temperature Gas-Cooled Reactor (HTGR) induces high acoustic sound pressure levels into the primary coolant (helium). This acoustic loading induces high cycle fatigue stresses which may control the design of the thermal barrier system. This study examines the dynamic response of a thermal barrier configuration consisting of a fibrous insulation compressed against the reactor vessel by a coverplate which is held in position by a central attachment fixture. The results of dynamic vibration analyses indicate the effect of the plate size and curvature and the attachment size on the response of the thermal barrier.

## NOMENCLATURE

A coverplate area, $m^2$. For a square coverplate, $A = L^2$.

D average diameter of central attachment fixture, m

E coverplate material elastic modulus, Pa

f natural frequency, $f = \omega/2\pi$, Hz

h thickness of coverplate, m

J area normalized acoustic coupling factor

$K_1$ design deflection and velocity amplification factor

$K_2$ design stress amplification factor, which is a function of coverplate - attachment fixture configuration

L length of one side of the coverplate, m

$P_p$ peak acoustical load, Pa

y maximum displacement of coverplate, m

$\dot{y}$ maximum velocity of coverplate, m/s

$\eta$ modal loss factor

$\lambda$ a dimensionless constant determined by analysis. Note that $\lambda$, $K_1$, $K_2$, and J are dimensionless factors which are a function of the geometry of the coverplate. There will be an infinite set of natural frequencies, which are indexed 1, 2, 3, ... corresponding to $f_1$, $f_2$, $f_3$ ... and $\lambda_1$, $\lambda_2$, $\lambda_3$ ... in ascending order.

$\nu$ Poisson's ratio

$\rho$ coverplate material mass density, $kg/m^3$

$\sigma$ maximum stress intensity in coverplate, the maximum difference in the principal stresses, Pa

$\omega$ radian natural frequency, rad/s

Subscript

R condition in reactor helium

## INTRODUCTION

This paper documents the response of a typical thermal barrier coverplate design to acoustic excitation. The rms octave band acoustic load in the HTGR-GT has been estimated at approximately 2000 Pa (160 dB) for frequencies between 22 and 716 Hz. This acoustic loading induces high cycle fatigue stresses, which may control the design of the thermal barrier system. This study examines the dynamic response of a thermal barrier configuration consisting of a fibrous insulation compressed against the reactor vessel by a coverplate, which is held in position by a central attachment fixture. Attention is directed to the response of the coverplate. It is recognized that the motion of the coverplate has the potential of damaging the fibrous insulation; however, the subject of fiber damage is not addressed in this paper.

## ACOUSTIC ANALYSIS THEORY

The response of a coverplate with a central fastener due to a random acoustic excitation can be estimated from Equations 1 through 4. These equations were derived based on the following assumptions: (1) thin flat plate theory applies; (2) the acoustic field is a narrow band random process (i.e., the plate resonance response is overlayed by an acoustic driver, which has a smaller bandwidth than that of the plate); and (3) the pressure field is uniform over the surface of the plate.

$$\dot{y} = \frac{K_1 P_p J}{\eta \rho h \omega} \tag{1}$$

$$y = \dot{y}_m / \omega \tag{2}$$

$$\sigma = \frac{12 K_2 K_1 J}{\lambda^4} \left(\frac{L}{h}\right)^2 \left(\frac{P_p}{\eta}\right) \tag{3}$$

$$\omega = 2\pi f = \frac{\lambda^2}{L^2} \left[\frac{Eh^2}{12\rho\,(1 - \nu^2)}\right]^{1/2} \tag{4}$$

Equation 3 implies that the maximum stress intensity in the plate is largely independent of the plate material. Once the four dimensionless factors ($K_1$, $K_2$, J, and $\lambda$) are determined for a given plate geometry, Equation 3 can be easily applied to predict the maximum stress intensity.

## COVERPLATE DYNAMIC MODEL

The nondimensional factors $K_1$, $K_2$, J, and $\lambda$ were computed for the coverplate geometry shown in Fig. 1. The coverplate is flat with the edge of the inner hole clamped and the outer edge free. There are 320 elements in the mesh. The material properties input into the MODSAP computer code (1) are the material density ($\rho$), elastic modulus (E), and Poisson's ratio ($\nu$) as shown Table 1. These factors are evaluated as a function of the ratio of the diameter of the central hole (D) to the side length of the coverplate (L).

The effect of plate curvature on the response of a coverplate was evaluated using the curved plate geometry shown in Fig. 2. The mesh and analysis procedures are identical to those for flat plates except, of course, that in the present case the plate is cylindrically curved. The plate was 1.905 cm (0.75 in.) thick and 50.8 cm (20 in.) on a side, with a central hole 11.7 cm (4.6 in.) in diameter. The material properties of the plate are indicated in Table 1. The outer edges were free and the plate was clamped along the edge of the central hole. Cylindrical curvature was introduced by curling two opposite sides of the plate upward out of the plane of the flat plate, as shown in Fig. 2. Thus, the curved plates all had the same projected area - the area of the shadow cast by a distant light source. The actual surface area of the plate and its mass increased with decreasing radii of curvature. The analysis was made mode by mode by loading the plate with an acceleration equivalent to a uniform pressure. The results were obtained for the first 20 modes.

### Plate Damping

The coverplate damping depends on a wide variety of parameters: (1) coverplate - attachment fixture configuration, (2) insulation material characteristics, (3) gas conditions, (4) frequency and mode shape of the plate, and (5) the coverplate material properties. It is very difficult to estimate the damping of a given system without testing. However, based on available data on similar types of thermal barrier

TABLE 1
MATERIAL PROPERTIES

| Property | Material (Metal) | A Typical Carbon-Carbon Composite | Fibrous Insulation, Saffil Alumina |
|---|---|---|---|
| Density, $\rho$ [$kg/m^3$ ($lb/in.^3$)] | 7916 (0.286) | 1716 (0.062) | 88 ($3.2 \times 10^{-3}$) |
| Elastic modulus, E [MPa (psi)] | $1.77 \times 10^5$ ($25.2 \times 10^6$) | $1.72 \times 10^4$ ($2.5 \times 10^6$) | $1.4 \times 10^{-2}$ (2.0) |
| Poisson's ratio, $\nu$ | 0.29 | 0.2 | --- |
| Allowable stress in high cycle fatigue, $S_a$[(a)] [MPa (psi)] | 66[(b)] ($9.5 \times 10^3$) | 27[(c)] ($3.9 \times 10^3$) | --- |

[(a)] For a design life of $1 \times 10^9$ s.
[(b)] For IN 713LC, which is a metallic material with high-temperature (900°C) properties sufficient to make it a top candidate material for the core outlet temperature (highest temperature) regions of a gas turbine nuclear reactor.
[(c)] Based on carbon-carbon with an expected average strength of 103 MPa (15,000 psi).

systems (2-4), it is seen that in atmospheric air $\eta > 0.02$ for low frequency ($f < 716$ Hz) excitation.

There are various theories for predicting plate damping at reactor operating conditions (i.e., helium at 70 bars and 850°C). In general, these theories indicate that the damping increases with gas pressure; however, there is no generally accepted theory. In spite of a lack of currently available test data, it is assumed in this study that at reactor conditions $\eta_R = 0.04$.

### Method of Determining Plate Response at Reactor Conditions

In general, the coverplate in a thermal barrier system is backed by fibrous insulation that exerts a force on the coverplate. However, the computer results generated in this study neglected this insulation force. This insulation, together with its interstitial gas, can cause a shift in the natural frequency of the plate. For typical thermal-barrier-size coverplates the frequency of the free coverplate is not significantly affected by the insulation in atmospheric air. However, there may be some small shift in the frequency at reactor (high pressure) helium conditions. If there is a shift in frequency, the plate stress ($\sigma$) and deflections (y) need to be corrected. This can be done by using Equation 5:

$$\sigma_R = \sigma \left(\frac{\omega}{\omega_R}\right)^2 \tag{5}$$

## RESULTS

### Flat Plate

Results were computed for the first 20 modes of six coverplate geometries corresponding to various ratios of the hole diameter D to the side length L (Fig. 1). The numerical values used in the input are nondimensionalized so that the results are applicable to any thickness or material of a thin coverplate of the form shown in Fig. 1. Results for $K_1$, $K_2$, J, $\lambda^2$, and the combined parameter $12K_1K_2J/\lambda^4$ are given in Tables 2 through 6. For the modes evaluated, the peak stresses occurred around the inner hole.

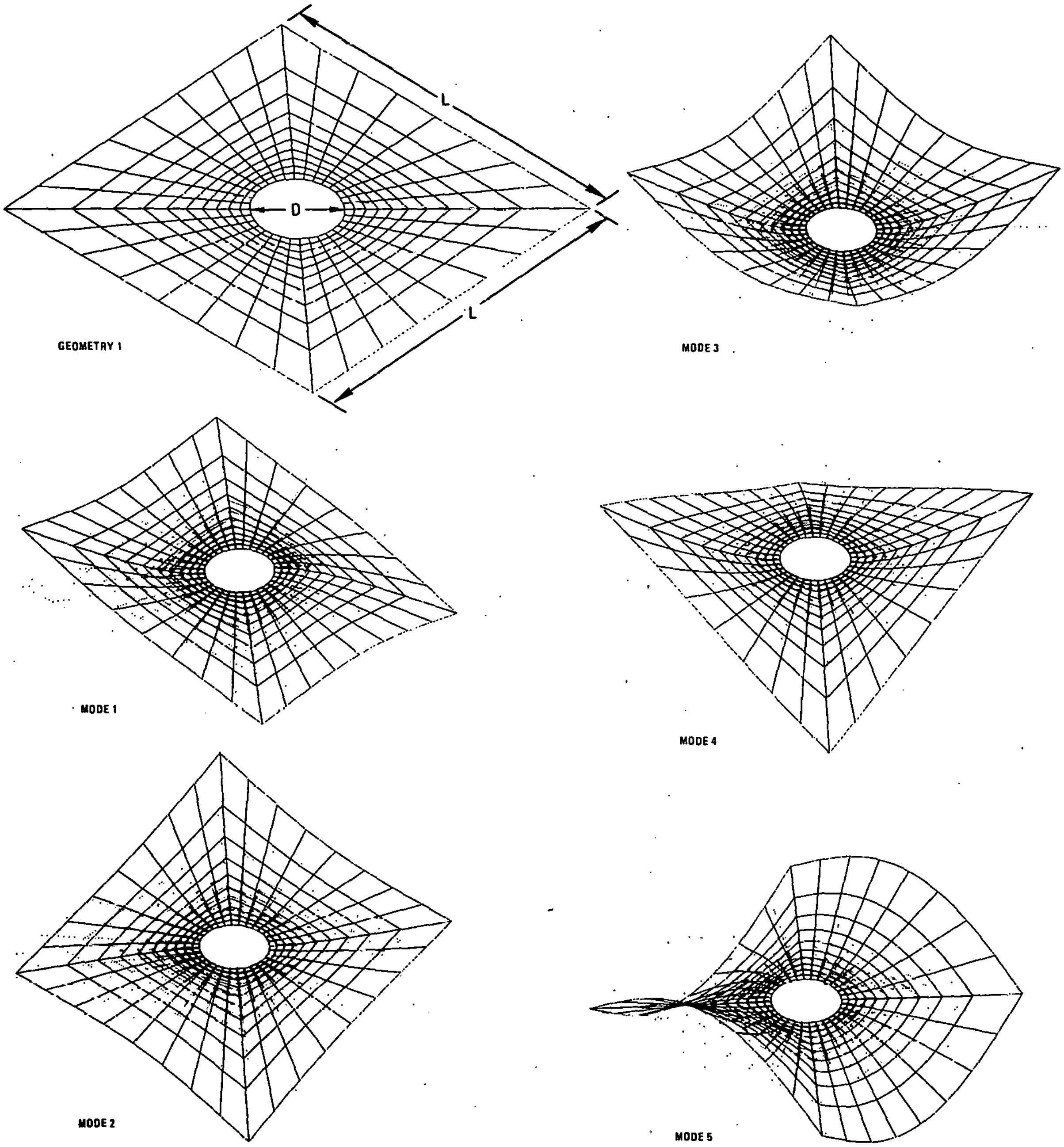


Fig. 1. Flat plate modes

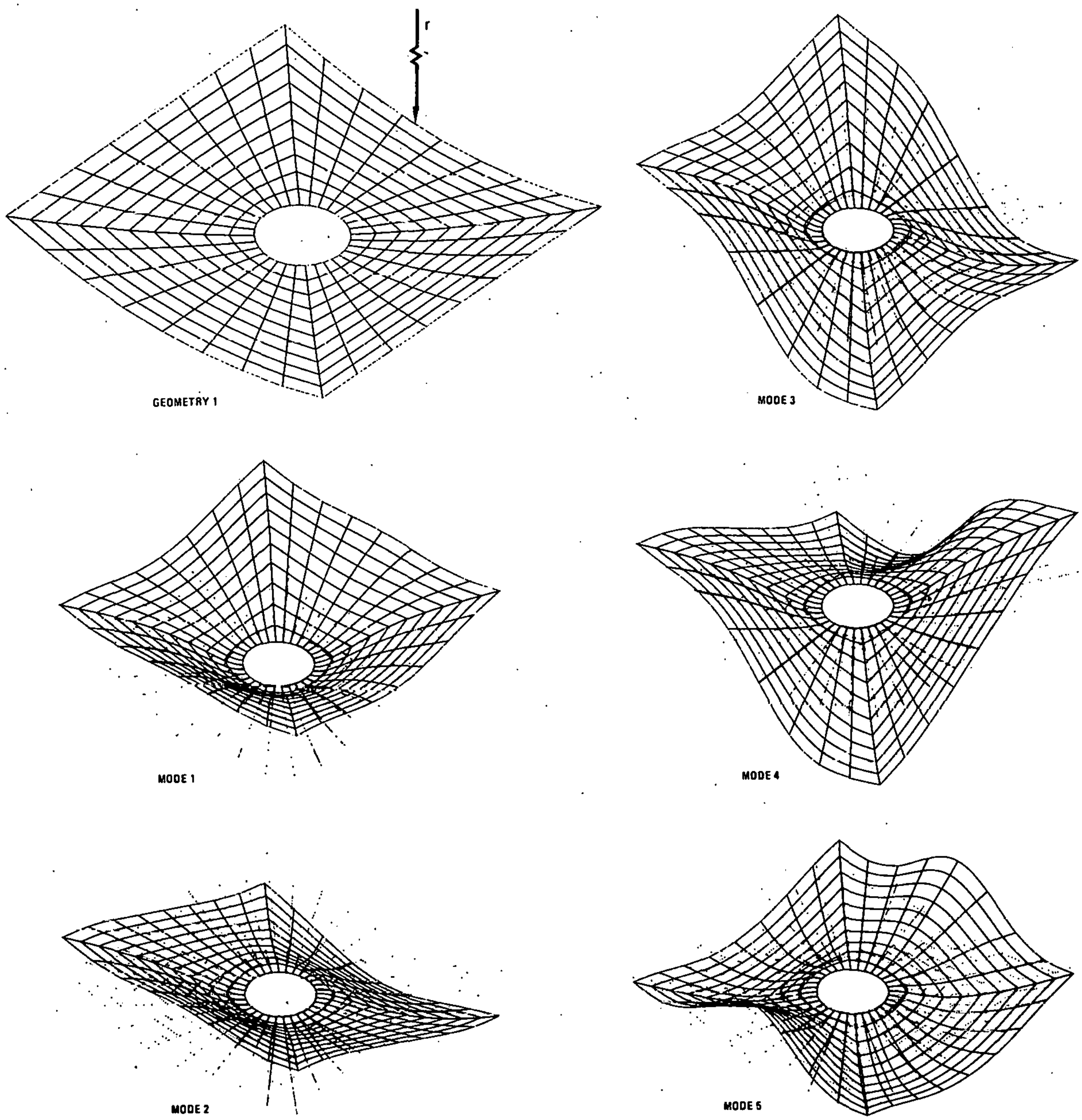


Fig. 2. curved plate modes. Radius of curvature = 86.4 cm.

TABLE 2
STRESS RESPONSE FACTOR, $12\ K_1 K_2 J/\lambda^4$

Hole Diameter/Side Length, D/L

| Mode | 0.1 | 0.2 | 0.23 | 0.3 | 0.5 | 0.7 |
|---|---|---|---|---|---|---|
| 1 | * | 0.1059 | * | * | * | * |
| 2 | * | * | * | * | * | * |
| 3 | 1.443 | 1.148 | 0.936 | 0.7565 | * | * |
| 4 | * | 0.0254 | * | * | 0.4123 | 0.1924 |
| 5 | * | * | * | * | * | * |
| 6 | * | * | * | * | * | * |
| 7 | * | * | * | * | * | * |
| 8 | $1.219 \times 10^{-1}$ | $1.093 \times 10^{-1}$ | $8.825 \times 10^{-2}$ | $8.400 \times 10^{-2}$ | $9.168 \times 10^{-2}$ | * |
| 9 | * | * | * | * | * | $5.701 \times 10^{-2}$ |
| 10 | * | * | * | * | * | * |
| 11 | * | * | * | * | * | * |
| 12 | * | * | * | * | * | * |
| 13 | $4.220 \times 10^{-2}$ | $3.185 \times 10^{-2}$ | $2.487 \times 10^{-2}$ | $1.886 \times 10^{-2}$ | $1.154 \times 10^{-2}$ | * |
| 14 | * | * | * | * | * | * |
| 15 | * | * | * | * | * | * |
| 16 | * | * | * | * | * | * |
| 17 | * | * | * | * | * | * |
| 18 | * | * | * | * | * | * |
| 19 | $1.151 \times 10^{-2}$ | $1.977 \times 10^{-2}$ | $2.082 \times 10^{-2}$ | $2.240 \times 10^{-2}$ | * | * |
| 20 | * | * | * | * | * | * |

*Calculated value was less than 0.01 which is not physically meaningful for design

TABLE 3
JOINT ACCEPTANCE $J$

Hole Diameter/Side Length, D/L

| Mode | 0.1 | 0.2 | 0.23 | 0.3 | 0.5 | 0.7 |
|---|---|---|---|---|---|---|
| 1 | * | $4.224 \times 10^{-2}$ | * | * | * | * |
| 2 | * | * | * | * | * | * |
| 3 | $8.721 \times 10^{-1}$ | $8.467 \times 10^{-1}$ | $8.394 \times 10^{-1}$ | $8.195 \times 10^{-1}$ | * | * |
| 4 | * | $2.159 \times 10^{-2}$ | * | * | $7.436 \times 10^{-1}$ | $6.357 \times 10^{-1}$ |
| 5 | * | * | * | * | * | * |
| 6 | * | * | * | * | * | * |
| 7 | * | * | * | * | * | * |
| 8 | $3.413 \times 10^{-1}$ | $3.482 \times 10^{-1}$ | $3.522 \times 10^{-1}$ | $3.642 \times 10^{-1}$ | $3.646 \times 10^{-1}$ | * |
| 9 | * | * | * | * | * | $5.878 \times 10^{-1}$ |
| 10 | * | * | * | * | * | * |
| 11 | * | * | * | * | * | * |
| 12 | * | * | * | * | * | * |
| 13 | $2.302 \times 10^{-1}$ | $2.261 \times 10^{-1}$ | $2.168 \times 10^{-1}$ | $1.891 \times 10^{-1}$ | $1.397 \times 10^{-1}$ | * |
| 14 | * | * | * | * | * | * |
| 15 | * | * | * | * | * | * |
| 16 | * | * | * | * | * | * |
| 17 | * | * | * | * | * | * |
| 18 | * | * | * | * | * | * |
| 19 | $1.234 \times 10^{-1}$ | $1.846 \times 10^{-1}$ | $2.091 \times 10^{-1}$ | $2.574 \times 10^{-1}$ | * | * |
| 20 | * | * | * | * | * | * |

*Calculated value was less than 0.01 which is not physically meaningful for design

TABLE 4
RESPONSE FACTOR $K_1$

Hole Diameter/Side Length, D/L

| Mode | 0.1 | 0.2 | 0.23 | 0.3 | 0.5 | 0.7 |
|---|---|---|---|---|---|---|
| 1 | 2.801 | 2.995 | 3.121 | 2.598 | 3.019 | 3.373 |
| 2 | 2.801 | 2.858 | 3.118 | 2.598 | 3.171 | 4.568 |
| 3 | 2.072 | 2.438 | 2.286 | 2.424 | 3.169 | 4.570 |
| 4 | 2.650 | 2.784 | 2.738 | 2.778 | 2.903 | 3.396 |
| 5 | 2.226 | 2.294 | 2.291 | 2.301 | 2.253 | 2.336 |
| 6 | 2.635 | 2.749 | 2.7699 | 2.672 | 2.980 | 3.006 |
| 7 | 2.635 | 2.748 | 2.769 | 2.671 | 2.981 | 3.006 |
| 8 | 3.241 | 3.276 | 3.236 | 3.143 | 2.605 | 2.534 |
| 9 | 3.452 | 2.560 | 2.532 | 2.526 | 2.499 | 2.611 |
| 10 | 3.452 | 4.047 | 4.144 | 3.301 | 3.329 | 4.094 |
| 11 | 2.536 | 3.984 | 4.145 | 3.302 | 3.379 | 4.094 |
| 12 | 3.360 | 3.631 | 3.572 | 3.680 | 3.811 | 3.625 |
| 13 | 2.736 | 2.792 | 2.717 | 2.531 | 3.314 | 3.269 |
| 14 | 2.935 | 2.603 | 2.787 | 3.038 | 3.552 | 3.679 |
| 15 | 2.936 | 2.477 | 2.787 | 3.038 | 3.552 | 3.679 |
| 16 | 2.037 | 2.609 | 2.2602 | 2.407 | 2.603 | 3.312 |
| 17 | 2.361 | 2.700 | 2.818 | 2.559 | 2.623 | 2.405 |
| 18 | 2.361 | 2.473 | 2.7801 | 2.559 | 3.09 | 3.471 |
| 19 | 3.004 | 3.198 | 2.995 | 2.704 | 3.093 | 3.471 |
| 20 | 2.080 | 2.231 | 2.1999 | 2.341 | 2.915 | 2.726 |

TABLE 5
STRESS FACTOR $K_2$

Hole Diameter/Side Length, D/L

| Mode | 0.1 | 0.2 | 0.23 | 0.3 | 0.5 | 0.7 |
|---|---|---|---|---|---|---|
| 1 | 11.15 | 11.30 | 9.678 | 10.67 | 11.15 | 14.94 |
| 2 | 11.16 | 10.40 | 9.679 | 10.67 | 11.09 | 14.87 |
| 3 | 9.645 | 9.021 | 8.8195 | 8.948 | 11.40 | 14.87 |
| 4 | 5.873 | 7.739 | 8.0904 | 8.830 | 10.76 | 14.71 |
| 5 | 13.08 | 20.60 | 18.61 | 21.26 | 34.32 | 50.11 |
| 6 | 17.17 | 22.07 | 19.35 | 21.07 | 33.85 | 49.94 |
| 7 | 17.17 | 18.83 | 19.67 | 21.09 | 33.85 | 49.93 |
| 8 | 18.07 | 19.23 | 16.60 | 18.33 | 36.28 | 43.05 |
| 9 | 27.86 | 18.79 | 18.86 | 18.93 | 22.81 | 63.31 |
| 10 | 27.86 | 26.21 | 22.79 | 22.39 | 28.86 | 42.67 |
| 11 | 18.62 | 23.52 | 22.79 | 22.39 | 28.85 | 42.67 |
| 12 | 17.45 | 21.44 | 21.52 | 22.17 | 23.84 | 37.44 |
| 13 | 44.35 | 42.80 | 38.24 | 40.42 | 32.51 | 57.22 |
| 14 | 58.00 | 45.41 | 33.62 | 32.70 | 39.56 | 63.81 |
| 15 | 57.99 | 41.53 | 33.62 | 32.71 | 39.57 | 63.81 |
| 16 | 55.31 | 57.09 | 55.13 | 47.83 | 45.17 | 74.27 |
| 17 | 56.05 | 103.4 | 78.19 | 99.95 | 50.66 | 77.60 |
| 18 | 56.06 | 95.22 | 79.28 | 99.97 | 51.86 | 74.29 |
| 19 | 41.03 | 50.49 | 53.77 | 68.65 | 51.84 | 74.29 |
| 20 | 59.38 | 73.60 | 72.29 | 70.72 | 58.48 | 81.95 |

TABLE 6
NATURAL FREQUENCY FACTOR $\lambda^2$

Hole Diameter/Side Length, D/L

| Mode | 0.1 | 0.2 | 0.23 | 0.3 | 0.5 | 0.7 |
|---|---|---|---|---|---|---|
| 1 | 9.806 | 12.72 | 13.79 | 16.23 | 25.60 | 43.56 |
| 2 | 9.806 | 12.82 | 13.78 | 16.23 | 25.76 | 44.03 |
| 3 | 12.04 | 13.95 | 14.73 | 16.78 | 25.76 | 44.03 |
| 4 | 13.47 | 14.82 | 15.42 | 17.16 | 26.00 | 44.51 |
| 5 | 20.20 | 22.89 | 24.20 | 27.93 | 48.66 | 106.88 |
| 6 | 34.75 | 36.65 | 37.50 | 39.88 | 56.30 | 109.45 |
| 7 | 34.75 | 36.73 | 37.50 | 54.75 | 73.80 | 109.45 |
| 8 | 44.36 | 49.07 | 50.72 | 54.75 | 73.80 | 113.26 |
| 9 | 63.07 | 66.82 | 67.18 | 68.15 | 76.24 | 139.43 |
| 10 | 63.07 | 68.61 | 70.64 | 75.47 | 94.96 | 147.83 |
| 11 | 65.75 | 68.73 | 70.64 | 75.47 | 94.96 | 147.83 |
| 12 | 66.70 | 71.35 | 73.26 | 78.69 | 102.83 | 155.23 |
| 13 | 89.30 | 100.9 | 104.26 | 111.2 | 134.72 | 198.98 |
| 14 | 108.31 | 116.8 | 118.33 | 121.8 | 139.43 | 201.90 |
| 15 | 108.31 | 116.8 | 118.33 | 121.8 | 139.4 | 201.90 |
| 16 | 108.85 | 117.3 | 120.47 | 127.6 | 147.6 | 207.56 |
| 17 | 118.69 | 127.5 | 134.90 | 152.3 | 197.6 | 253.11 |
| 18 | 118.39 | 129.0 | 134.90 | 152.3 | 210.6 | 273.73 |
| 19 | 125.90 | 134.7 | 139.31 | 154.5 | 210.6 | 273.73 |
| 20 | 139.67 | 150.6 | 155.05 | 167.1 | 223.8 | 297.16 |

Example

Consider a square coverplate 50.8 cm (20 in.) on a side and 1.90 cm (0.75 in.) thick with a 11.7 cm (4.6 in.) diameter hole. The parameters describing the plate are:

$$L = 50.8 \text{ cm}, D = 11.7 \text{ cm}, h = 1.9 \text{ cm}, D/L = 0.23$$

The combined stress response parameter is read from Table 2. This parameter is at its maximum in the third mode.

$$\frac{12K_1K_2J}{\lambda^4} = 0.9360$$

The maximum stress in the plate can now be found directly from Equation 3:

For $P_p$ = 2216 Pa and $\eta$ = 0.04,

$$\sigma = 0.9360 \left(\frac{20}{0.75}\right)^2 \frac{1}{0.04} \times 2216 \text{ Pa}$$

$$\sigma = 37 \text{ MPa (5400 psi)}$$

The remaining nondimensional parameters in the third mode are:

$$K_1 = 2.286 \qquad \lambda^2 = 14.73$$

$$K_1 = 8.8195 \qquad J = 0.8394$$

The natural frequency of the plate in the third mode is given by Equation 4 with the material properties indicated in Table 1.

$$f \text{ (Hz)} = \begin{cases} 161.6 \text{ Hz, carbon-carbon} \\ 243.6 \text{ Hz, steel} \end{cases}$$

The maximum displacement can similarly be calculated from Equation 2.

Table 7 summarizes the response of selected sizes of flat plates with integral attachment fixtures. If the maximum modal stress $\sigma_R$ in Table 7 is compared to the allowable stress ($S_a$) (per Table 1), it is seen that, for a metal plate made of IN 713LC, most of the sizes selected are within the allowable stress in the plate. However, a plate of similar size made of a carbon-carbon (C-C) composite will have stresses above the allowable stress for that material.

Table 7 was generated based on the simplified assumption that the maximum plate stress can be calculated based only on the most severe mode. This implies there is no need to combine modal stresses. This is justified based on the fact that the stress parameter in Table 2 for this most severe mode is at least an order of magnitude larger than for any other mode, and it has been assumed that $P_p/\eta$ is independent of frequency. If this latter assumption is not true, it would be necessary to combine modal stresses.

Curved Plates

The results for the mode with the highest stress for each radius of curvature are given in Table 8. It can be seen from Table 8 that curving the plate:

1. Increases the natural frequency.
2. Decreases the modal stress.
3. Decreases the modal displacement.

Curving the plate substantially increases the plate stiffness, raising both the natural frequencies and the resistance of the plate to respond to all dynamic loads. As with the flat plate, the peak stresses occur around the inner hole.

The mode shapes of a flat plate and the plate with r = 86 cm (34 in.) are compared in Figs. 1 and 2. The first two modes of the flat plate are missing in the curved plate. However, the higher modes are very similar. Both the maximum stress and maximum displacement were found always in all cases to arise in

TABLE 7

SUMMARY OF MAXIMUM STRESS IN A FLAT COVERPLATE WITH AN INTEGRATED CENTRAL ATTACHMENT FIXTURE

(Based on Eqs. 3 and 5, Table 2, and $\eta_R$ = 0.04)

| Area, $A = L^2$ [$m^2$ ($in.^2$)] | Thickness, h [cm (in.)] | D/L | $\frac{12K_1K_2J}{\lambda^4}$ Per Table 2 | Frequency, f(a) (Hz) | | Maximum Plate Stress, $\sigma_R$(b) [MPa (ksi)] | Maximum Plate Velocity, $\dot{y}$(c) [cm/s (in./s)] | |
|---|---|---|---|---|---|---|---|---|
| | | | | Metal | C-C | | Metal | C-C |
| 0.372 (576) | 1.9 (0.75) | 0.192 | 1.172 | 172 | 112 | 66.4 (9.83) | 76 (30) | 540 (210) |
| 0.258 (400) | 1.9 (0.75) | 0.23 | 0.936 | 247 | 161 | 36.8 (5.34) | 46 (18) | 330 (130) |
| 0.258 (576) | 2.54 (1.0) | 0.23 | 0.936 | 330 | 215 | 20.7 (3.0) | 25 (10) | 180 (72) |
| 0.372 (576) | 2.54 (1.0) | 0.192 | 1.172 | 229 | 149 | 37.4 (5.42) | 43 (17) | 310 (120) |
| 0.258 (400) | 3.8 (1.5) | 0.23 | 0.936 | -- | 99 | 9.21 (1.34) | | 83 (33) |

(a) Based on Eq. 4 with material properties per Table 1 and $\lambda$ as given in Table 6.

(b) Based on $P_p$ = 2216 Pa (0.321 psi) and assuming $f/f_R = 1$.

$$\sigma_R = \frac{12K_1K_2J}{\lambda^4} \left(\frac{L}{h}\right)^2 \frac{P_p}{0.04}$$

(c) Combining Eqs. 3, 4, and 1 yields

$$\dot{y} = \sigma_R \left(\frac{\rho E\, 12}{1 - \nu^2}\right)^{-1/2} \left(\frac{\lambda^2}{K_2}\right)$$

TABLE 8
CHARACTERISTIC OF MODE WITH GREATEST RESPONSE TO A PRESSURE FIELD

| Radius of Curvature, r(a) [m (in.)] | Mode | Ratio of Natural Frequencies, $f_r/f_{r=\infty}$(b) | Ratio of Maximum Modal Stress, $\sigma_r/\sigma_{r=\infty}$(c) | Ratio of Maximum Modal Displacement, $y_r/y_{r=\infty}$ | Dimensionless Factors | | | |
|---|---|---|---|---|---|---|---|---|
| | | | | | J | $K_1$ | $K_2$ | $\lambda^2$ |
| ∞ (flat) | 3 | 1.0 | 1.0 | 1.0 | 0.8394 | 2.286 | 8.09 | 14.73 |
| 5.38 (212) | 2 | 1.15 | 0.93 | 0.72 | 0.8579 | 2.12 | 10.4 | 16.94 |
| 2.03 (80) | 2 | 1.50 | 0.73 | 0.35 | 0.8733 | 1.75 | 16.9 | 22.10 |
| 1.02 (40) | 1 | 2.07 | 0.51 | 0.20 | 0.9048 | 1.75 | 21.4 | 30.49 |
| 0.864 (34) | 1 | 2.26 | 0.46 | 0.17 | 0.9128 | 1.75 | 22.8 | 33.29 |
| 0.508 (20) | 1 | 3.06 | 0.34 | 0.087 | 0.9372 | 1.67 | 31.6 | 45.07 |

(a) The results were generated for metallic plates with a projected area of a 50.8 cm x 50.8 cm flat plate (i.e., A = 0.258 $m^2$), a thickness of 1.90 cm, and with a central hole 11.7 cm in diameter.

(b) $f_r$ = natural frequency of the mode with the maximum stress for a coverplate with a radius of curvature r

$f_{r=\infty}$ = natural frequency of the mode with the maximum stress for a flat (r=∞) coverplate

(c) $\sigma_r$ = maximum modal stress for a given radius of curvature r

$\sigma_{r=\infty}$ = maximum modal stress for a flat plate (r=∞)

the umbrella-like mode (mode 1 for r = 86 cm and mode 3 for the flat plate). This mode also had the highest area normalized acoustic coupling factor J.

No attempt was made to reduce the results further as was done for the flat plate since thin plate theory is not applicable to curved plates and the available theories for cylindrically curved panels do not permit simple generalization of the finite element results.

## CONCLUSIONS AND RECOMMENDATIONS

A set of equations has been given that is useful for predicting the dynamic response of a flat cover-plate. The equations show that the maximum stress intensity in a plate is largely independent of the plate material. Based on the work done to date, a reasonable size (per Table 7) thermal barrier cover-plate can withstand the low frequency design acoustic loading currently predicted for the HTGR-GT plant.

Based on the present study of the response of cylindrically curved coverplates to acoustic loads it is found that the introduction of curvature reduces the response of a coverplate. The natural frequencies of a coverplate increase with increasing curvature and both the maximum stress and maximum displacement fall with increasing curvature. It is recommended that the general method for flat plates be applied to curved coverplates since it yields a conservative estimate of the plate response. If additional margin is required, credit for curvature can be obtained only from a case by case study such as is presented in this paper.

Before a given coverplate - attachment fixture can be designed with confidence in an HKGR-GT, it is necessary to obtain a more precise definition in the following areas:

1. Acousting loading: Need to increase confidence in the acoustic load as a function of frequency and location in the primary coolant loop.
2. Material structural properties: Need to determine the high cycle fatigue strength of the candidate high-temperature materials in reactor helium. Currently there are only limited data on materials capable of withstanding the design temperatures in the core outlet region of an HTGR-GT.
3. Plate damping: Need to determine experimentally the damping characteristics in reactor helium. Currently the plate damping in reactor helium is based on limited testing of similar structures and on theory, which needs verification.
4. Loading on attachment fixtures: The results of this study do not include the loading on the attachment fixture; this information is needed.
5. Verification of plate boundary conditions.
6. The effect of the plate motion on fiber damage needs to be determined experimentally.


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