STRUCTURAL INTEGRITY OF A GAS TURBINE COMBUSTION SYSTEM
SUBJECTED TO INCREASED DYNAMIC PRESSURE

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ABSTRACT

The effect of combustion dynamic pressure oscillations on the structural integrity of the MS 7001F dry low NOx 2 (DLN 2) combustion system has been evaluated using ANSYS [ref. 1] finite element analyses and high cycle fatigue material data. Analytical results were validated with laboratory measurements on the combustion system subjected to combustion dynamic pressure at actual gas turbine temperature and pressure operating conditions. The combustion liner, transition piece, impingement sleeve and supports were proven to have excellent durability when subjected to dynamic loads. No risk of structural failure exists at anticipated dynamic pressures using assumptions shown to be conservative.

INTRODUCTION

The DLN 2 combustion system includes the liner and transition piece assemblies which contain and direct the high temperature combustion gases. These components are restrained at minimal locations and with adequate flexibility to allow for thermal expansion. The combustion process, especially premixed burning, can have pressure oscillations at discrete frequencies due to flame stability. This stimulus combined with flexible attachments results in considerable mechanical vibration. This vibration can cause fatigue cracking which could lead to catastrophic failure.

Mechanical development of new combustion components and systems must trade off adequate flexibility to minimize thermal stress with sufficient stiffness to avoid vibration problems. Thermal stresses are routinely calculated for component design. However, vibratory stresses have not previously been adequately quantified. Structural dynamic analyses had not been utilized due to the complexity of the geometry, loading and contact behavior. Measurements of strain were also of limited usefulness because critical locations are unknown and often inaccessible and also due to the high metal temperatures. Previous structural dynamic analysis of a simplified combustion model with components represented by rigid linkages and corresponding laboratory testing demonstrated that the system actively responds to the dynamic pressure [ref. 2]. However, no information regarding component vibratory stresses could be obtained from such analyses.

Historically, allowable limits on combustion dynamic pressure for structural integrity were based on field experience gained from years of operation. Direct application of field experience from different combustion systems to new designs was questionable. Analytical prediction of component life and development of a design methodology which considers dynamic pressure loading was therefore undertaken to assure the integrity of mechanical designs.

Nonlinear transient dynamic finite element analyses were necessary to predict the combustion system dynamic behavior which accurately compared with measured strains and accelerations. Component stiffnesses and masses as well as distributed pressure loading could then be accurately modeled. Contact behavior including the sliding friction at supports and seals was essential to computing the response. These numerous sliding interfaces add considerable damping to the system. Nonlinear gap conditions capable of maintaining or breaking physical contact according to relative displacements between components were also necessary. Attempts to utilize viscous damping and no gap condition to eliminate nonlinearities did not yield accurate results.

Modal analyses were used to compute natural frequencies and mode shapes. These results compared well with laboratory modal testing and were used to determine the proximity to resonance conditions.

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The finite element model in Figure 1 was constructed with particularly accurate representations in the areas of highest structural concern. These areas focus on the supports and regions of past structural problems. Accurate representation means geometry is to drawing, fillets are included and element size, shape and aspect ratios are very good. This level of detail captures stress concentrations with the forced response analyses but would not be required for the modal analyses. Mesh density was initially based on past work with some refinement added as analyses were performed. Nearly all 8 noded brick elements were used to minimize the number of elements for what was expected to be a large model and a computationally intensive solution. Geometric symmetry was not exploited because future work may investigate effects of nonsymmetric loading and constraints. All results presented here are symmetric with respect to an axial-radial plane.

The transition piece assembly has accurate representations of the aft mount, body and forward sleeve assembly. The aft frame has accurate stiffness and mass but the rib fillets are not included because these are near the neutral axis of the frame and therefore have low stress. The frame also lacks cooling holes. The impingement sleeve is accurate except for the lack of cooling holes. Stress concentration factors can be applied if large vibratory stresses occur near these holes.

The liner has accurate representations of the support stops, stiffening ribs and cooling ring details except for the cooling holes. Again stress concentration factors can be applied if large vibratory stresses occur near the holes. Nodes around the liner align with the forward sleeve for point to point attachment of the hula seal.

The hula seals on the forward and aft ends of the liner are modeled as a combination of a normal spring and a tangential friction element. The hula seal has a stiffness computed with a small finite element model and a friction coefficient as measured in hula seal fretting tests.

Several notable details of the combustion system have not been included in the model. Seals at the aft end of the transition...
piece to the turbine section and adjacent transition pieces have not been modeled. Uncertainty exists regarding the contact force of these seals with the transition piece due to relative thermal expansion of the components. Due to the relatively small mass and stiffness of these seals, the only anticipated effect on the dynamic response is a reduction in damping. Therefore, this assumption is conservative. Crossfire tubes which connect between adjacent combustion liners were not included. The connection of these relatively small tubes to the liner is very flexible so that minimal influence on the system behavior is expected. Also, because the crossfire tubes contact two different liners responding to dynamic pressure with some unknown phase difference, inclusion in this one combustion chamber model would be difficult.

The components modeled are considered to be far more flexible than the supporting structure which is considered ground in this model.

**NONLINEAR TRANSIENT DYNAMIC ANALYSES**

The transient dynamic finite element analysis determines the dynamic response of a structure to time dependent loads. The basic equation of motion is solved using finite element representations for the stiffness, mass and damping of the structure. The direct integration method is used to solve for the unknown displacements at discrete time points. Numerous solutions are obtained at successive time points to represent the pressure oscillations.

The nonlinearities modeled are gap conditions and Coulomb friction. The huia seals at the liner forward end and between the liner and transition piece are prestressed, can maintain or break physical contact depending on relative displacements and stick or slide tangentially with Coulomb friction. Gap conditions and Coulomb friction are also modeled on the liner stops and forward supports and on the aft mount to bracket which also includes preload due to the bolt torque.

The pressure loading represents measurements from the laboratory while simulating actual machine conditions. Peak to peak pressure of 2 MPa at 160 Hz defines the combustion dynamics. Pressure measurements were made at several locations in the combustion system. The distribution of pressure was then predicted based on the theoretical shape of the dynamic combustion pressure wave.

System damping was specified using Rayleigh mass and stiffness damping constants. Ratios of actual to critical damping were measured for each individual component. Mass and stiffness constants were then calculated. Sensitivity of the response to the damping constants were performed which showed very small effect.

Super-elements were utilized to reduce computer run times. This technique was well suited to this problem because linear portions of the model (super-elements) are interconnected at relatively few points by the nonlinear contact elements.

**LABORATORY VALIDATION**

Strains and accelerations were measured in the laboratory at numerous locations. Figure 2 plots the computed oscillating axial displacement at an accelerometer location on the transition piece forward end. The ratio of dynamic displacement to the static displacement at 2 MPa dynamic pressure is 0.14. Integrating the measured acceleration at this location assuming a sinusoidal response yields a ratio of 0.16. The analysis has under predicted displacement at this location by 12%. The location with the poorest correlation, radial motion at the aft end inner panel of the transition piece, had computed displacements 5 times greater than measured. In this case, the analysis predicted vibration of the inner panel of the transition piece. One possible explanation for this over prediction is that this vibration is actually damped by the seals which are not included in the model. Subsequent wear and differences in component alignment during manufacturing could change this damping, so assuming no damping is conservative.

Circumferential vibratory strain divided by allowable strain between the aft mount and aft frame is plotted for several elements in Figure 3. The time range in this plot is for 2 cycles and the dynamic pressure is 2 MPa. These elements cover an axial length of approximately 10 mm. A strain gage approximately centered in this region measured a ratio of 0.0038. Similarly good correlation exists at all gages with the exception of the gages associated with the inner panel vibration discussed above. Also, because the strain gages were positioned prior to completion of the analyses, half of the gages measured extremely low strain levels as predicted by the analysis. Future testing can have much better gage locations based on the analytical results.

![Figure 2. Computed transition piece axial displacement versus time at forward end.](https://proceedings.asmedigitalcollection.asme.org/doi/abs/10.1115/1.3486392)
Figure 3. Computed circumferential strain versus time for several elements located between the aft mount and aft frame on the centerline of the transition piece.

HIGH CYCLE FATIGUE LIFE PREDICTION

High cycle fatigue is a function of vibratory stress, mean stress, metal temperature and material type. Material testing was initiated before the vibratory stresses were computed but the mean stress and metal temperatures were known. Therefore, testing was limited to maximum mean stresses and metal temperature for each material over a range of vibratory stresses. Although the critical location may be at a lower mean stress and metal temperature, this material testing program determines a conservative limit for vibratory stress. Allowable dynamic strain was then established.

Critical locations were determined by computing dynamic strain range, the metal temperature and mean stress at each element. Figure 4 is a plot of equivalent vibratory strain divided by allowable strain at one critical location. The strain is far below the allowable level. Note that measuring strain at this location would not be feasible demonstrating the necessity for relying on analytical methods.

IMPACT LOADING

The nonlinear transient dynamic analysis also predicted the response to the initial impact of the dynamic pressure. The pressure was conservatively assumed to begin oscillating at the full amplitude rather than ramping up as is probably more realistic. The highest dynamic amplification in the system occurs at the transition piece forward support (Figure 5). The gap condition at this support is alternating between contacting and being free resulting in nearly a factor of 12 increase over the static reaction at impact. Fortunately, the support areas are designed for higher impact load based on historical design practices.
MODAL ANALYSES

In addition to predicting structural dynamic response for the particular frequency of interest, a wider frequency range was investigated to assure that slight variation in stimulus frequency would not be significant. Modal analyses were more expeditious than the nonlinear transient dynamic for evaluating frequency effects.

Experimental measurement of natural frequencies was performed to validate the finite element analyses. The testing was limited to individual components in a free support condition at room temperature.

The analytical prediction of natural frequencies agrees very well with the measurements as do the mode shapes which were compared by viewing animated displays. The variation of ± 7% is partly due to differences in mass and stiffness from manufacturing tolerances as well as measurement inaccuracies. The conclusion from this validation is that finite element analyses can be used with confidence to predict natural frequencies for actual support conditions and metal temperatures.

Modal analyses results for the combustion system with actual support conditions at operating metal temperature are given in Figure 6. The combustion system consists of the liner and transition piece assemblies with radial spring elements for the hula seals. The hula seals are preloaded to represent the assembly interference and thermal expansion but do not develop force due to axial sliding friction. Results without hula seal prestress is also given. The liner stops are restrained in the axial and circumferential directions relative to the axis of liner. The transition piece forward supports are restrained in the circumferential direction relative to the axis of liner. The aft bracket is fixed at the connection to the first stage nozzle outer support ring. The bolted connection between the aft mount boss and aft bracket which can actually slide after overcoming static friction is modeled as both fixed and pinned, ie. free to rotate, to test sensitivity since the actual condition can not be represented.

<table>
<thead>
<tr>
<th>Mode</th>
<th>Fixed Brkt Freq (Hz)</th>
<th>Free Brkt Freq (Hz)</th>
<th>No Seal Prestress Freq (Hz)</th>
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Figure 6. Computed natural frequencies of combustion system.

The conclusion from these analyses is that several natural frequencies are very close to the combustion dynamic pressure, i.e. the third and fourth modes. The response to mode three was captured with the nonlinear transient dynamic analyses and did not cause structural concerns. Mode four is a liner natural frequency which was analyzed separately and found to be of no consequence. The response is not extremely sensitive to the aft bracket rotation condition except for mode 3. Most importantly, no additional modes correspond to the combustion stimulus frequency due to variation in this boundary condition. The hula seal prestress also has minimal effect, especially in the frequency range of interest.

CONCLUSIONS

The analyses predict a large margin of safety against loss of structural integrity at possible dynamic pressure levels. Higher dynamic pressure than expected has also been analyzed to estimate the operational pressure limit with still no integrity concern. Modal analyses were also performed to be aware of nearby resonances.

The successful demonstration of this capability provides a powerful tool for gas turbine combustion development. Areas where the first signs of fatigue would most likely occur have been identified. Future strain measurements can be made at better locations as indicated by the analyses. Reactions including initial impact exerted on supporting components were determined which are being used to design these components. Relative motion at interfaces were also computed and can be used to understand wear at these locations.

The analysis has several conservative assumptions which caused over prediction of some measured accelerations and strains. Future work will be to include more interface components and the resulting friction, verification of the dynamic pressure distribution and additional measurements of the component behavior.

REFERENCES