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A System Dynamics Approach to Modeling Gas Turbine Combustor Wear

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ABSTRACT

This paper deals with the development of the basic understanding of dynamic and wear problems of combustor systems. A system dynamic modeling approach has been developed using a general purpose dynamic/wear analysis program (Mechanism Analysis Program, *MAP*), and excellent validation has been achieved via laboratory tests. The objective is to develop a generic design/analysis tool, both, for quick evaluation of preliminary designs, and for detailed design studies to create improved/optimized combustor designs from a wear/life perspective.

Dynamic simulations of two different designs are run: Many worn field samples are available for one design, and laboratory test results for the other. The wear patterns on the former are matched very well by the analytical model. Excellent correlation, too, is obtained with the dynamic (accelerometer) measurements on the latter. It is, now, possible to isolate system dynamic causes (such as resonance) from material related ones (such as hardness) when designing for long life.

1. INTRODUCTION

The GE MS9001E (MS9E) heavy-duty gas turbine is a packaged power plant for electric utility applications or as an industrial power generation unit (Favreau, 1984). The turbine cross-section is shown in Figure 1. The MS9E has a single shaft with a speed of 3000 rpm providing direct drive to a 50 Hz generator. The base load firing temperature is $1085^oC(1985^oF)$. In simple-cycle, the MS9E has a base load rating of 109.9 MW and a 32 percent thermal efficiency (LHV) when burning distillate fuel at ISO conditions of $15^oC(59^oF)$ and 1.013 bar (14.7 psia, 101.5 kPa). The reverse-flow combustion system shown in Figure 2 utilizes fourteen 356 mm (14 inches) diameter slot cooled combustion liners oriented at a cant angle of 13 degrees to the machine axis. Compressor discharge air flowing at 885 lb/sec (402 kg/sec), $342^oC(648^oF)$ and 11.9 bar (173 psia, 1.2 MPa) enters the liners through combustion, dilution and cooling holes. In the liner, gas or oil fuel is injected and burned; oil at a rate of 17.45 lb/sec (7.92 kg/sec). In addition, water or steam can be injected to reduce pollutants and/or augment power output. Combustion zone gas temperatures exceed $1650^oC(3000^oF)$. The combustion gas flows into the transition piece (TP) shown in Figure 3 where it turns parallel to the machine axis and transitions from a circular cross-section to the annular first stage nozzle opening.

Mechanical design of the combustion system is extremely challenging due to the combination of high metal temperatures and thermal gradients, high pressure and combustion driven pressure oscillations,large aerodynamic reactions and sealing requirement between components. The fundamental design approach followed in the MS9E is to have minimal restraint on each component, to allow free thermal expansion while resisting the aerodynamic and gravitational loads. Thus, the liner has three supports located at 120 degree spacing around the fuel nozzle end to resist the combustion pressure drop forcing

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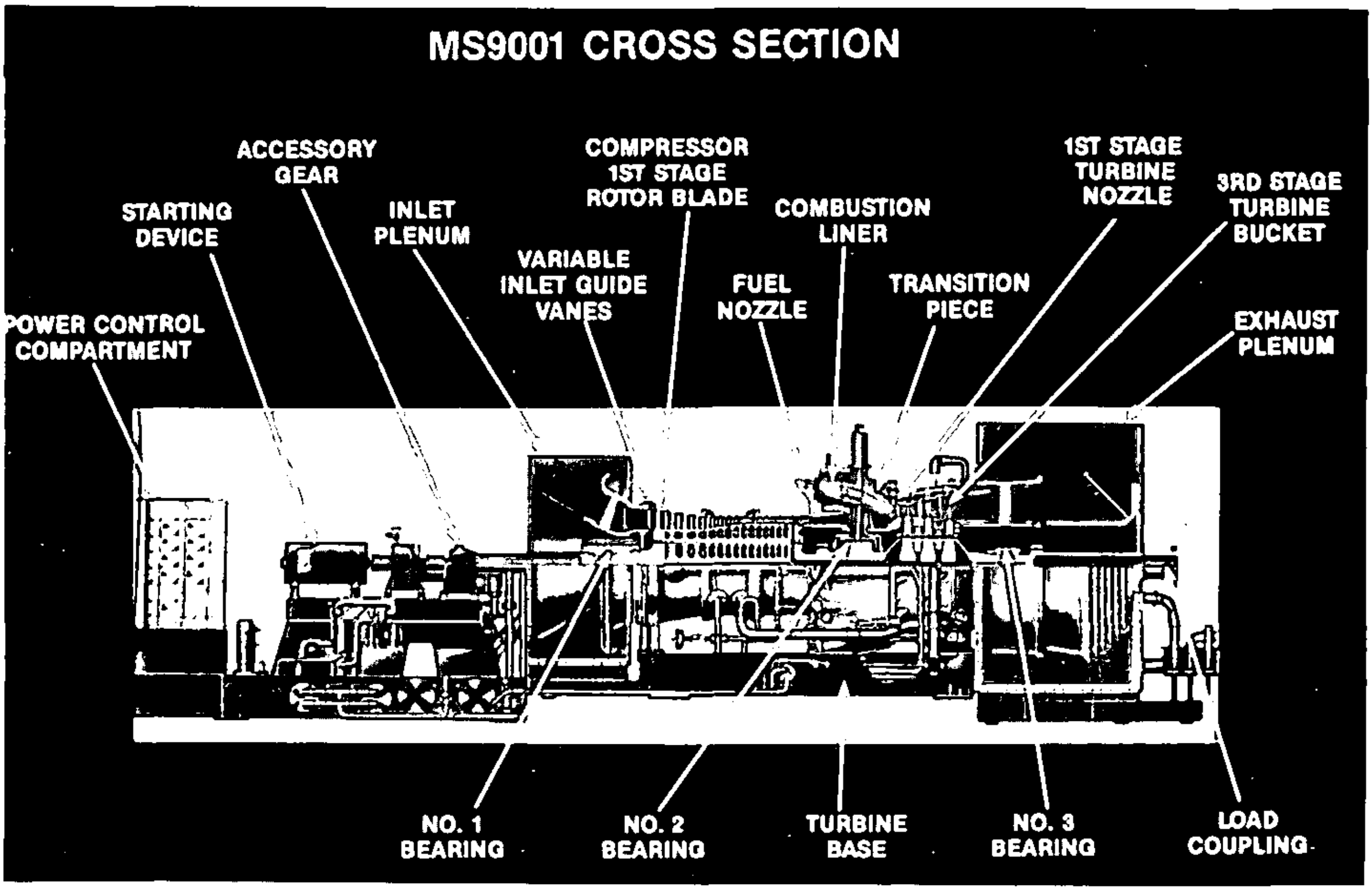


Figure 1. Cross Section of the MS9001E Gas Turbine

the liner downstream. The liner can then thermally expand in diameter and in length. Likewise, the TP aft mount provides axial, radial and circumferential support to oppose aerodynamic force resulting from changing the combustion gas flow direction and velocity. Only radial and lateral support is provided at the TP forward end to allow free thermal expansion. The downstream end of the liner fits inside the TP contacting with a spring "hula" seal which allows the two parts to slide relative to one another. Between the TP and the first stage nozzle are inner and outer "floating" seals which allow differential motion. Adjacent TP's have side seals to stop leakage. Other seal/water interfaces include the fuel nozzle to liner and cross-fire tube to liner junctions which are also designed to allow differential motion.

The MS9001E-Uprate (MS9E-U) is a new gas turbine offering increased performance over the standard machine. The firing temperature has been increased to $1124^oC (2055^oF)$. Changes to the combustion system are seen in a 381 mm (15inches) longer TP, relocated TP forward supports to the inner radius of the machine and increased liner wall thickness.

2. TECHNICAL APPROACH

System Dynamics Modeling:

A typical combustor is a multibody system with moving masses, springs, contact surfaces, etc. Wear at all contact surfaces is affected by the dynamic behavior of the entire system, and small changes to any one part influences the whole system, and hence the wear at all the contact surfaces. Simple wear theories could easily characterize the energy lost in rubbing at these surfaces, and their influence in the system differential equations. The contact surfaces themselves can reasonably be represented as between spheres or, as between hollow cylinders and spheres. All these can be incorporated into a general purpose multibody dynamic analysis code such as *MAP*. The interplay between the system dynamics and wear are represented in the chart in Figure 4.

MAP is a general computer-aided design and analysis tool capable of analyzing 3-D mechanical systems comprised of rigid and/or flexible members with relative motion constraints. These systems can be analyzed for equilibrium configurations, or may be in motion. Motion of moving systems may be fully specified (kinematic), or may be the outcome of

REVERSE FLOW COMBUSTION SYSTEM

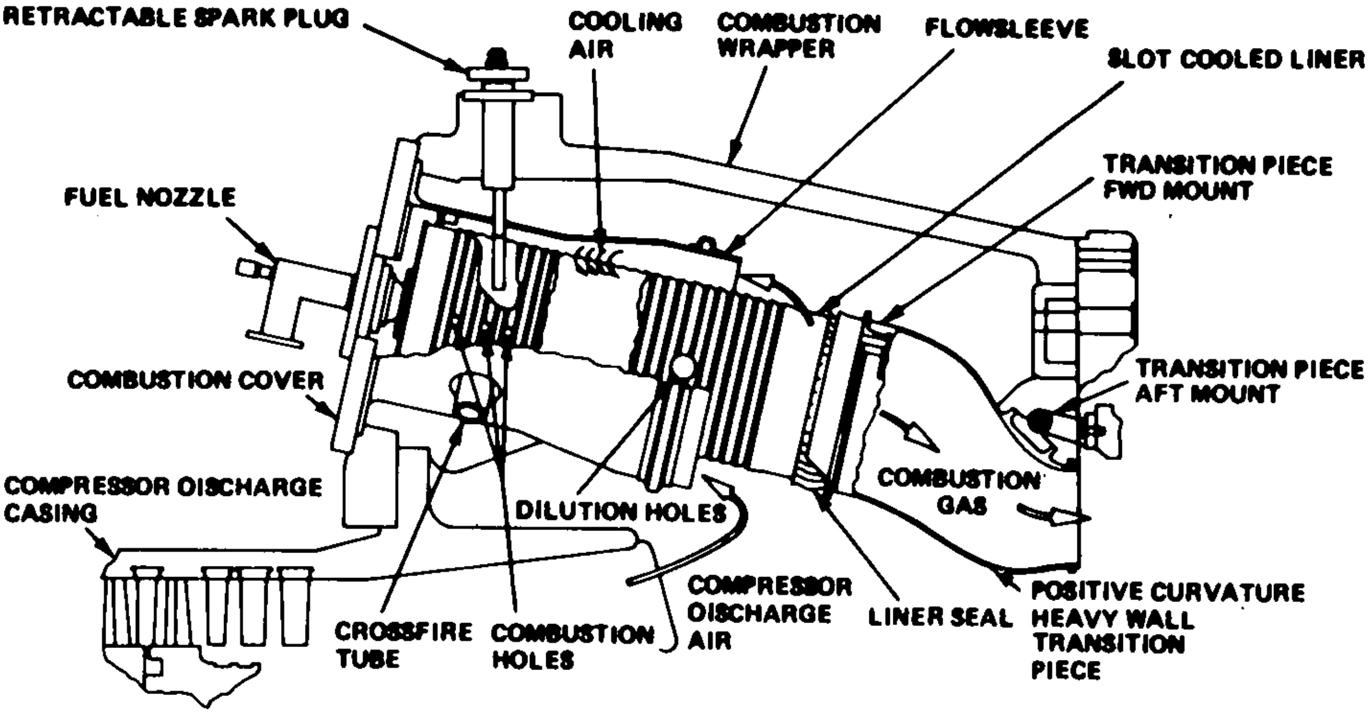


Figure 2. Combustion System of the MS9001E Gas Turbine

loading conditions to which the system is subjected (dynamic). *MAP* automatically formulates the governing differential equations of motion of the mechanical system and solves them (MAP Manual, 1991). In the context of the current work, *MAP* is used as a tool to simulate the contact loads and rubbing velocities of all combustor components by Bagepalli et al.(1991).

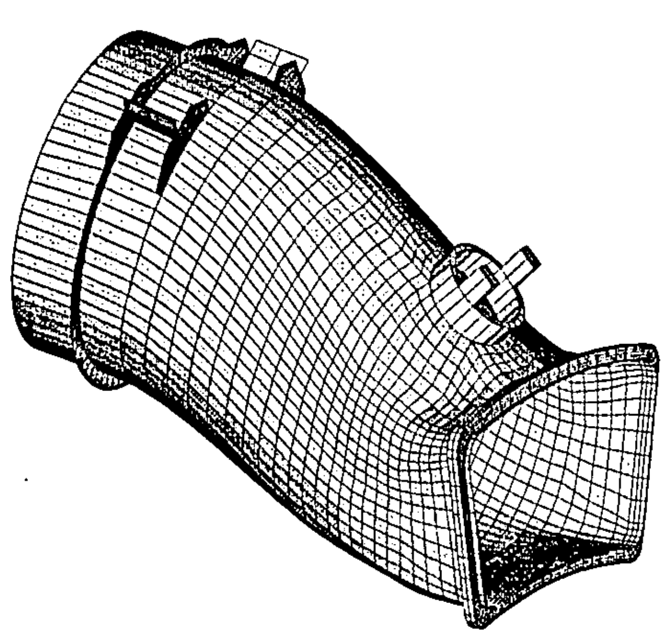


Figure 3. Finite Element Model of the Transition Piece (TP)

The main thrust behind this work can be summarized in the following points:

- To develop a basic understanding of the combustor system dynamics and wear problem, and to perform a root-cause analysis to isolate the most critical design and operational parameters.
- To develop a generic design/analysis tool, both, for quick evaluation of preliminary designs, and for detailed design studies.
- To develop improved/optimized combustor designs from a wear/life perspective.

Initially, the MS9E combustor is analyzed, since it is in service at various sites and field data is available. Later, the MS9E-U design is modelled. This design has only been lab-tested, and therefore served as a validation tool of the analytical approach. Comparisons between the two designs are shown later.

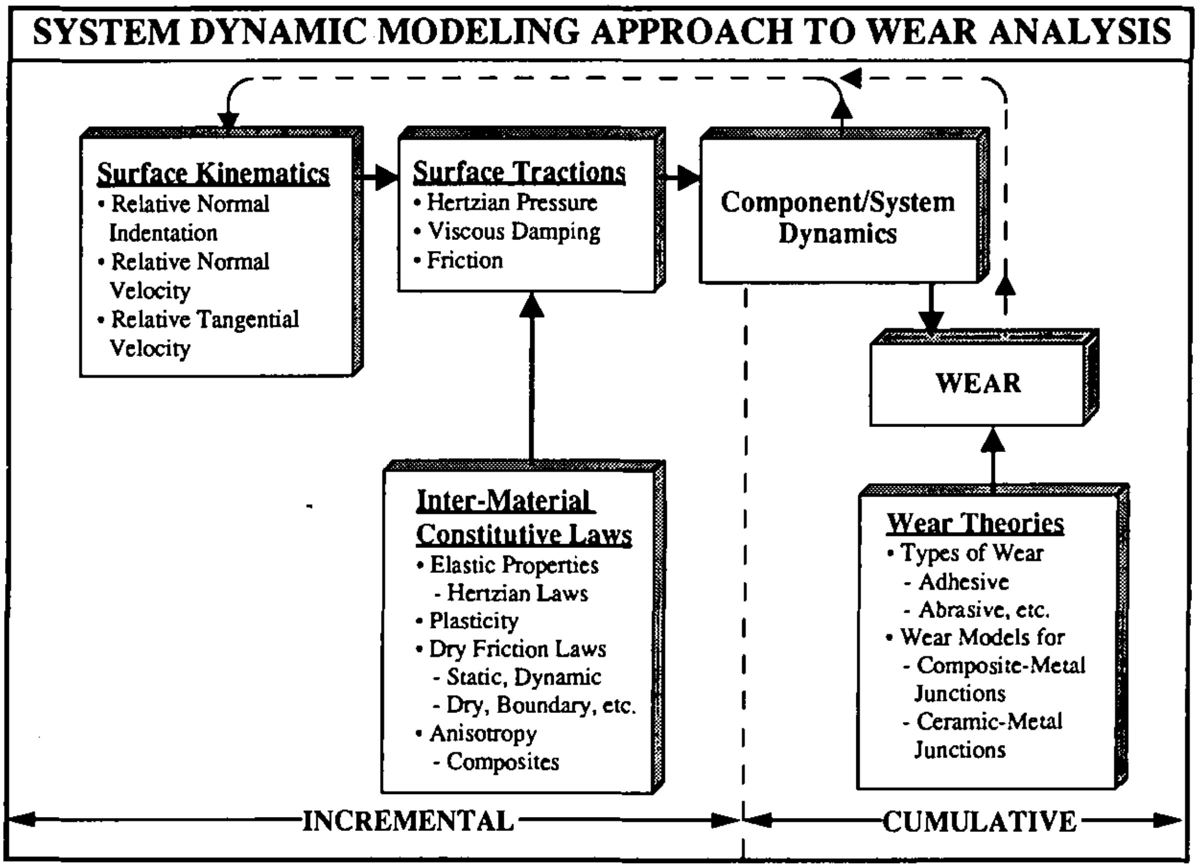


Figure 4. System Dynamic Modeling Approach to Wear Analysis

3. MODELING

The first step is to develop an analytical micro-dynamic model of the entire combustor unit shown in Figure 5. The following is a list of items to be considered:

1. Identify the relevant geometric properties of the non-moving members of the combustor system.
2. Identify the mass, inertia and relevant geometric properties of the moving members of the combustor system:
   - Liner:
     MS9E Combustor:
     Weight = 44.8 lbs (20.3kg)
     $Inertias = I_{xx}(1980.0)$, $I_{yy}(15544.0), I_{zz}(15544.0) lb\text{-}in^2$.
     MS9E-Uprate Combustor:
     Weight = 69 lbs (31.3 kg)
     $Inertias = I_{xx}(3167.0), I_{yy}(19558.0)$, $I_{zz}(20454.0), I_{xy}(168.0) lb\text{-}in^2$
   - Transition Piece (TP):
     MS9E Combustor:
     Weight = 82.0 lbs (37.2 kg)
     $Inertias = I_{xx}(7663.0), I_{yy}(15492.0)$, $I_{zz}(18675.0), I_{xy}(-2375.0) lb\text{-}in^2$
     MS9E-Uprate Combustor:
     Weight = 117.9 lbs (53.5 kg)
     $Inertias = I_{xx}(14791.0), I_{yy}(21052.0)$, $I_{zz}(29042.0), I_{xy}(2067.0) lb\text{-}in^2$
   - Inner Floating Seal:
     Weight = 1.0 lb (0.45 kg)
     $Inertias = I_{xx}(25.1), I_{yy}(10.1)$, $I_{zz}(0.5), I_{xy}(-0.07) lb\text{-}in^2$
   - Outer Floating Seal:
     Weight = 1.2 lbs (0.54 kg)
     $Inertias = I_{xx}(43.5), I_{yy}(17.6)$, $I_{zz}(0.8), I_{xy}(-0.08) lb\text{-}in^2$
3. Represent the "mobility" (ie., degrees of freedom) of the identified members, including freedoms necessary to model misalignment. This is done, typically, by a series of concatenated relative degrees of freedom (typically, via pin, slider, planar, spherical, cylindrical and universal joints) between bodies to achieve the required total mobility. It is important to note, here, that in order to represent the local structural flexibility at the transition piece aft mount area (both on the support side and on the TP side), the aft mount was given a longitudinal and lat-

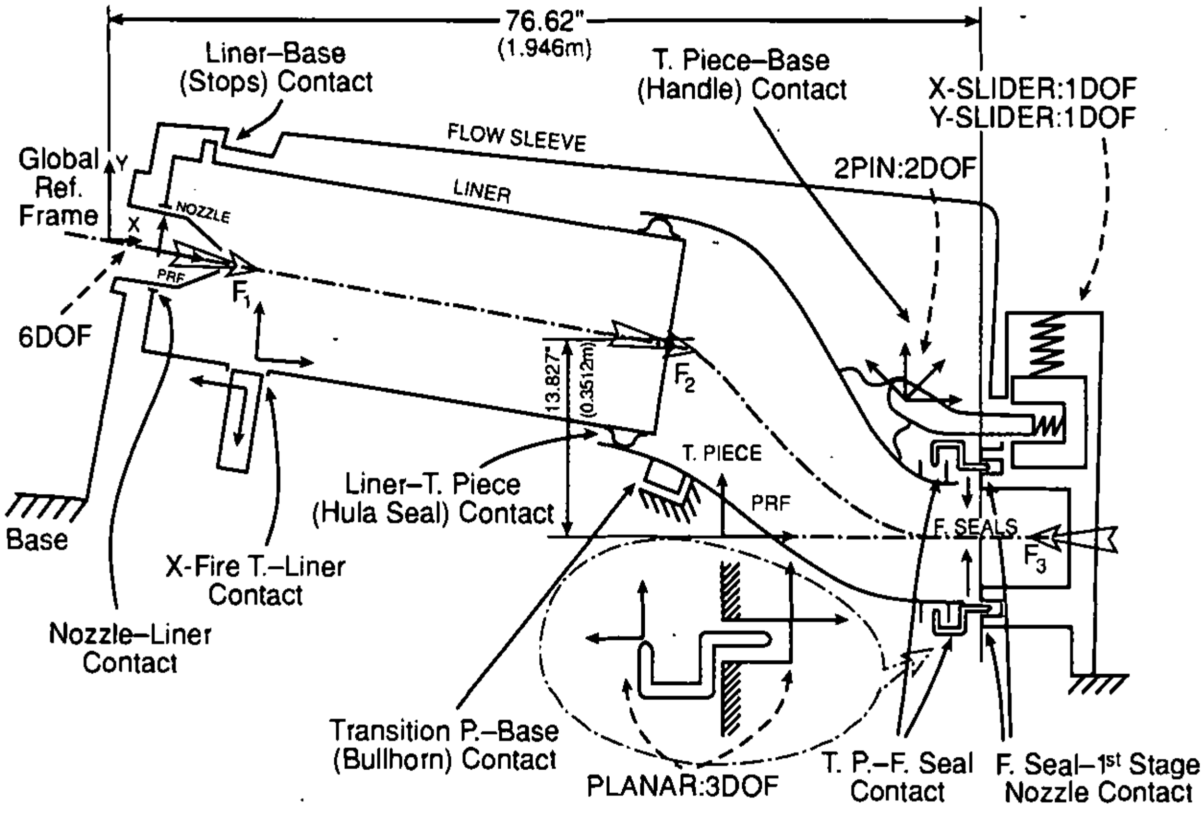


Figure 5. Combustor in a Ground-Based Gas Turbine

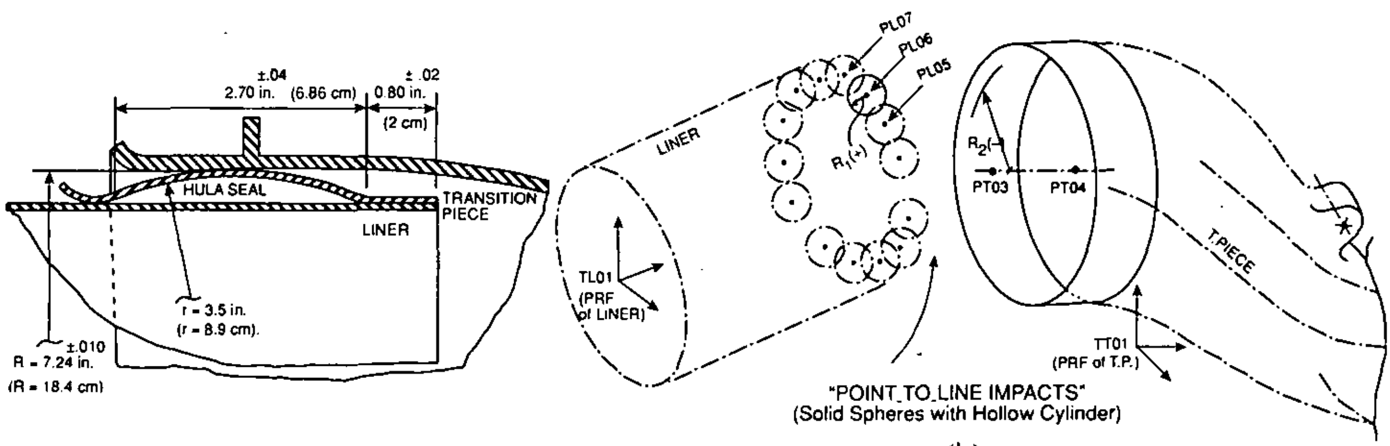
eral stiffness via sliding joints with appropriate springs and dampers (whose stiffnesses were determined from a separate FEM analysis of a TP assembly).

4. Represent all junctions, stops and contacts via appropriate models for all "inter-member" rubbing or contact situations. This factors in:

   - Complex time-varying kinematics of contact - geometries, relative position and velocity vectors represented, either in closed form, or via finite element (surface/boundary) nodes and elements.
   - Traction theories for inter-material rubbing: Constitutive laws for Coulomb friction, contact stresses/pressures - including dependencies on temperature, velocity, wear, and other parameters.
   - Wear theories: analytical models appropriate to each pair of members at all junctions, including the effects of orthotropy (for wear of composites): Adhesive wear, Delamination wear, Abrasive wear, etc.

There are numerous contact/junction situations in the MS9E and the MS9E-U combustor systems. All contact situations are briefly summarized below. A detailed description of the hula seal contact is given later as an example.

- Fuel-nozzle/Liner contact: It is represented as contact between a hollow cylinder and (the tops of) two closely placed spheres. Appropriate internal and external radii are chosen to represent the proper tolerances between the two bodies.
- Liner forward stops: There are three liner stops at the nozzle end of the liner. These comprise slotted brackets welded on to the liner surface which ride over fixed blocks. Relative travel (both, fore-aft and lateral) between the slotted brackets and the blocks is limited. The fore-aft stop is modeled via three small spheres (attached to the liner) rubbing against the surface of a very large sphere (representing a flat surface).
- Liner lateral stops: These are part of the stops identified above and are modelled via three large spheres rubbing against a central sphere.
- Liner/Transition-Piece (TP) contact or, the *Hula Seal*: The 44 clip-like fingers contacting the inside of the transition piece are, each, modeled as individual sphere to cylin-



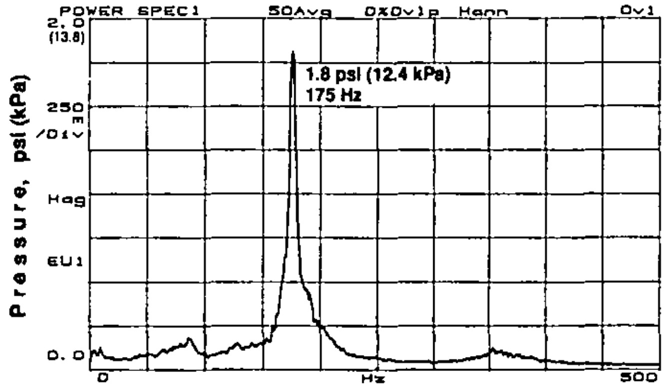
(a) (b)

Figure 6. Model of the Liner-Hula Seal-Transition Piece Contact **a)** Real System **b)** Presentation in the *MAP*

der (hollow) contact as shown in Figure 6. The contact stiffness used is the experimentally measured value (or, alternatively, as evaluated from a separate FEM analysis). There will be about 37.5 lbs (167 N) preload on each hula seal finger when Liner-Hula Seal-TP are assembled.

- Outer Floating Seal (OFS) / TP junction: This and the remaining three junctions (below) are represented in the following way: given the relative motion between each floating seal and an adjacent body (transition piece, or, the first-stage nozzle), the rubbing that *can* occur between the two bodies is broken down into several contact subsystems of edges rubbing against faces. When modelling the edge of a lip rubbing against the face of a flange, the face is represented as a line (or generic cylinder) and the edge as a point (or a generic small sphere).
- OFS/First-stage Nozzle junction, see above.
- Inner Floating Seal (IFS)/TP junction, see above.
- IFS/First-stage Nozzle junction, see above.

5. Add all springs and dampers in the system, including those needed to represent attachment flexibility, such as the aft mount attachment: these are $K_x = 65,700 lb/in (11,520 kN/m)$ and $K_y = 11,500 lb/in (2,016 kN/m)$.
6. Include any joint friction, such as the *Break Away Torque* at the aft mount hinge: this is experimentally measured to be 840 inch-pounds (95 N-m).
7. Add the appropriate gravity field vector. Based on field samples the most severe cases were the combustors located at the *bottom* of the gas turbine unit, ie., with the combustors pointing *upwards*, these have gravity pulling *down* on them which translates into gravity acting *upwards* in the *MAP* model.
8. Input proper operating conditions: pressures, temperatures, loads etc. The total pressure inside the combustor unit is lower than outside. This pressure comprises a static pressure and a dynamic (acoustic) pressure, the latter being represented as a sum of harmonic (sinusoidal)



(a)

$$F_1 = F_{1static} + F_{1dynamic} * Sin(\omega t)$$

$$F_2 = F_{2static} + F_{2dynamic} * Sin(\omega t - \psi)$$

$$F_3 = F_{3static} + F_{3dynamic} * Sin(\omega t - \xi)$$

(b)

Figure 7. **a)** A Typical Dynamic Pressure Measurement at the Fuel Nozzle End **b)** The Loads on the Liner and Transition Piece

pulses. The pressure loading on the combustor unit is input at three stages along the combustor column: at the fuel nozzle end, F_1 acting on the liner; at the liner/transition-piece (tp) interface, F_2 acting on the tp; and at the first stage nozzle end of the transition piece, F_3 also acting on the tp as shown in Figure 5. These loads have two components each; static load which is steady-state aerodynamic load, and the dynamic load which is obtained from a pressure transduser located at the fuel nozzle end. The dynamic loads at the liner-tp interface and at the first stage nozzle are obtained by suitably modifiying the pressure at the nozzle end by considering area ratios and plus a phase shift (ψ and ξ). Figure 7 show a typical dynamic pressure measurement (peak-to-peak 1.8 psi(12.4 kPa) at 175 Hz) at the fuel nozzle end and the loads on the liner and transition piece.

4. RESULTS

Wear Calculations:

After a proper *MAP* model is developed, dynamic simulations of the combustor system yield time-histories of different system variables such as contact loads, relative rubbing velocities, indentations, etc. Simple wear models are constructed to evaluate the amount of wear and degradation that can occur in such a system. In order to properly model the wear at each contact, one must first have a good knowledge of the *type* of wear occurring: for example, whether Abrasive, Delamination, Adhesive, or other. Typically, one uses the *Archard* equation for Adhesive/Fretting type of wear (Archard, 1956):

$$\dot{W} = K\frac{LV}{3H}\delta t \qquad (1)$$

where, $\dot{W}$ is the wear rate ($mm^3/second$), L and V are respectively the instantaneous contact load and the rubbing velocity, H is the hardness, and δt is the time increment, K is the wear coefficient which depends on the rubbing material pair, type of friction (dry, lubricated, etc.), and type of wear (adhesive, abrasive, fretting, etc.). In this case, L and V are given by *MAP* Analysis results for each contact, total time t is taken from MS9E inspection manual for each element, H is picked from hardness-temperature charts as a function of real body temperatures on different locations of the combustor. The total wear in a given time t, is obtained by multiplying the time-averaged wear-rate by this time. This rate is given by:

$$\dot{W}_t = \frac{1}{t_{max}}\int_0^{t_{max}} \dot{W} \qquad (2)$$

Evaluation of the wear coefficient K is based on scanning electron microscopy (SEM) photographs of each piece and laboratory reports, and wear coupon tests under different conditions (loads, speeds, temperatures up to 1450^oF) for many years. The wear coefficient K is determined to be approximately 5×10^{-4}. This is confirmed by literature (Rabinowicz, 1980, 1981).

Now, a dynamic simulation of the entire system is run and one can map on to all contact surfaces their respective cumulative wear profiles. If test results are available, one can iterate on the model to match exact wear patterns.

Some Wear Results:

Here we will briefly compare some field measurements with analysis results at different junctions, but some detail is given about the hula seal contact as an example.

- Fuel nozzle wear: Nozzle/Liner contact; Based on the field inspection manual, the maximum acceptable wear depth is 0.06 inches (1.52 mm) over 90 degree arc (about 4 inches in length). Getting the normal load and rubbing velocity (L*V) on the contact surface,using Eq.1, and noting (from worn fuel nozzle samples) a wear width of about 0.50 inches (12.7 mm), the wear depth in 8000 hours (which is the present inspection interval) is 0.07 inches (1.78 mm) from the *MAP* model which is very close to the acceptable wear depth.
- Hula seal wear: Liner/TP contact; There are 44 fingers on the hula seal, and each sees a distinct rubbing velocity and load. Despite the apparent symmetry in the combustor system, the wear patterns observed on the hula seals were skewed, suggesting that this might be due to some misalignment. The additional degrees of freedom

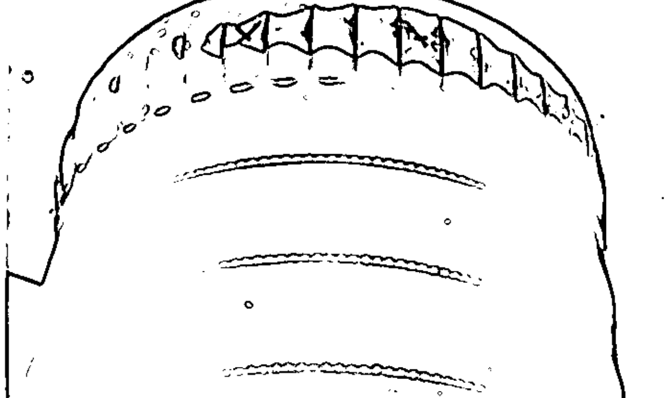


Figure 8. Worn Hula Seal Field Sample

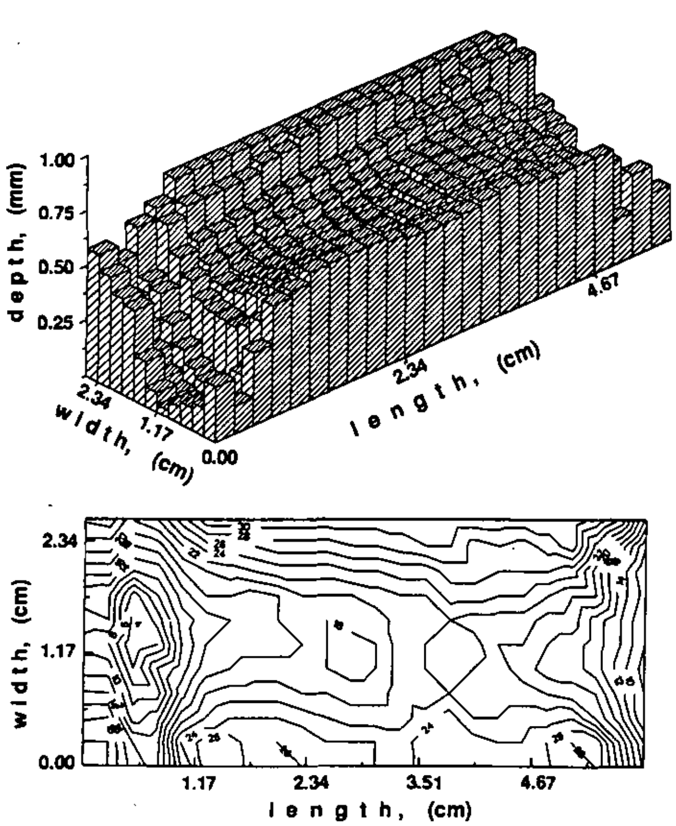


Figure 9. Wear Pattern on the Hula-Seal Finger #1

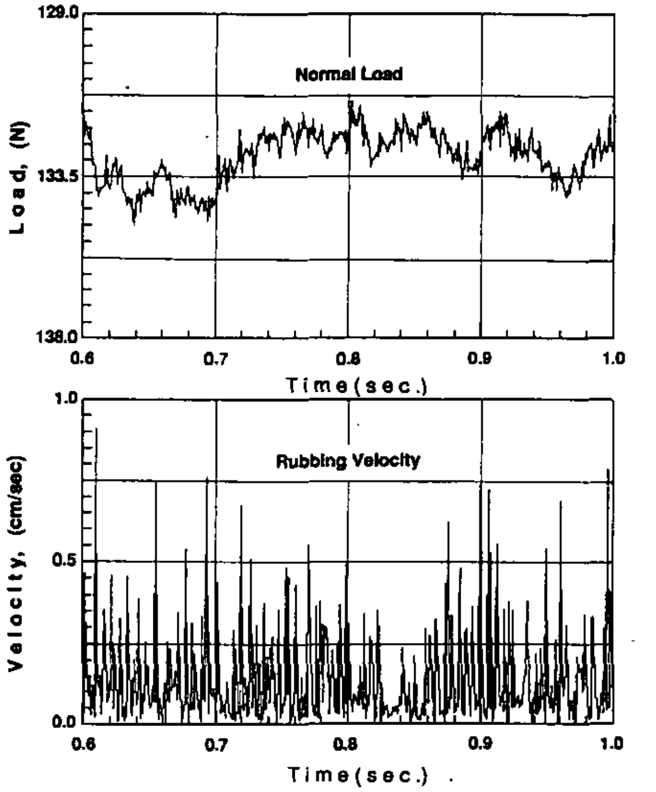


Figure 10. Normal Load and Rubbing Velocity on the Finger #1

due to misalignment (to the idealized 2D model) needed to be properly represented. Some worn hula field samples were available and a worn hula seal field sample is shown in Figure 8. Detailed wear measurements were made on all hula seal fingers. Figure 9 shows a contour plot and 3D plot of wear pattern of finger #1. The total wear measured on each of the forty-four fingers is shown in Figure 11a. Normal loads and rubbing velocities $(L * V)$ on each finger are given by MAP analysis program, as an example Figure 10 shows normal load and rubbing velocity for finger #1. The wear coefficient, K is calculated from wear data on coupon tests, and is about 4.8×10^{-4}. Analysis results are subsitituted into Eq.1 and calculated wear volumes are shown in Figure 11 for each finger by using same amount of service hours of field samples. Measured wear volumes are within 95% of the analytical volumes. Some fingers have worn beyond their inspection limit which says "wear diamond patterns must not touch". Another observation of the hula seal wear patterns, the fingers see longitudinal rubbing at the top and bottom, and

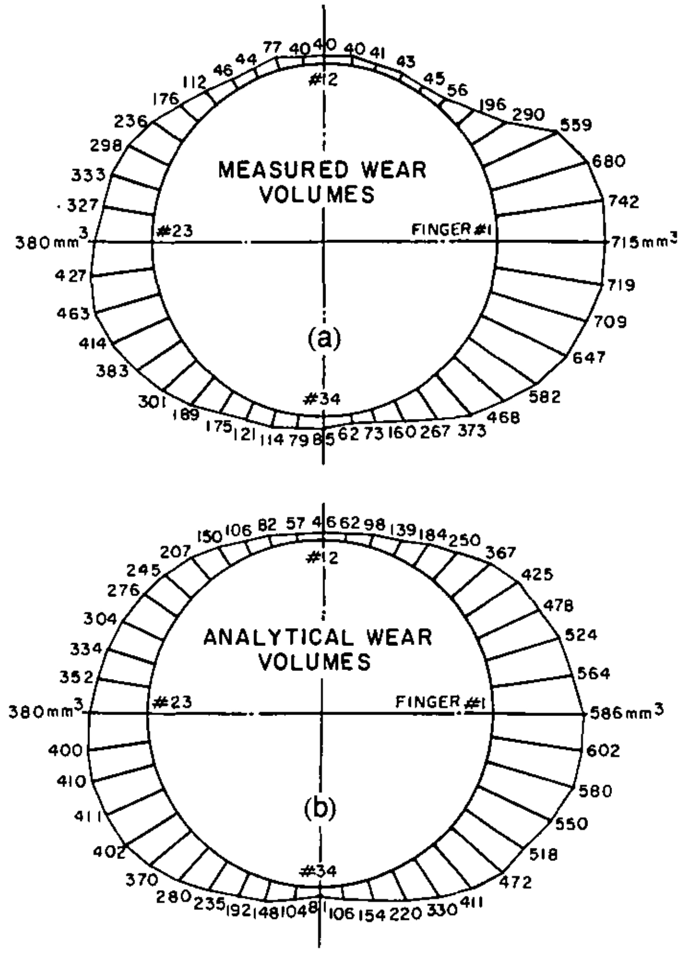


Figure 11. a) Measured, and b) Predicted Wear Volumes of Forty-four Hula-Seal Fingers

lateral rubbing at the sides. The analytically obtained relative velocity vectors confirmed these findings.

- Liner-Stops Wear, Cross-Fire Tube Wear, Transition Piece-Forward Mount Wear: Results are presented by Bagepalli et al.(1991) with all details.

- Floating Seal Wear: Floating Seals sit between the TP and First-Stage Nozzle,at the aft end of combustor and function as sealing elements. Outer, Inner, and Side Seals rub against, both, the TP and the First-stage Nozzle. There are several edges rubbing against several faces. Using analysis results $(L * V)$, for example, total wear depth calculated for 8000 hours of service (for a contact area of 0.2 × 7.5 inches) is 0.034 inches (0.86 mm); this is very close to the allowable 0.031 inches (0.79 mm) wear depth for same service hours.

MS9E Gas Turbine has 14 combustors. These results are given for the top combustor (12 o'clock position). The *MAP* model is then modified to be able to simulate the other combustors. The fuel nozzle and hula seal wear results are given in Figure 12 as a field measurement and analysis result. The dark area on the small circles shows fuel nozzle wear, and hula seal wear is shown on the bigger circle and categorized as severe, mild, or light wear for each combustor. Analysis results match with field measurements very well (Figure 11 and Figure 12). The same type of analysis was performed on the MS9E-U. The predicted wear volume on each component is compared with those of the MS9E combustor.

MS9E-U Test Combustor:

MS9E-U is a new design machine and has no field wear measurements. Some acceleration measurements are done at GE Power Generation labs on the MS9E-U and analysis results are matched with those measurements.

Some Accelerometer Results: The MS9E-U design is modeled in a way similar to the MS9E: The TP is longer and heavier, and the forward mounts are on the inner radius. The (pressure) loads applied are as measured in the test. The test configuration had twelve accelerometers located (six on the Liner, and six on the TP) - see Figure 13. From the dynamic simulation we are able to evaluate the accelerations at these same locations and directions. Figure 14 shows a time-plot of these accelerations and corresponding analytical results. Figure 15 and Figure 16 show comparisons between some test and analytical acceleration results in the frequency domain from liner and transition piece respectively. Wear calculations are also done on all contacting surfaces

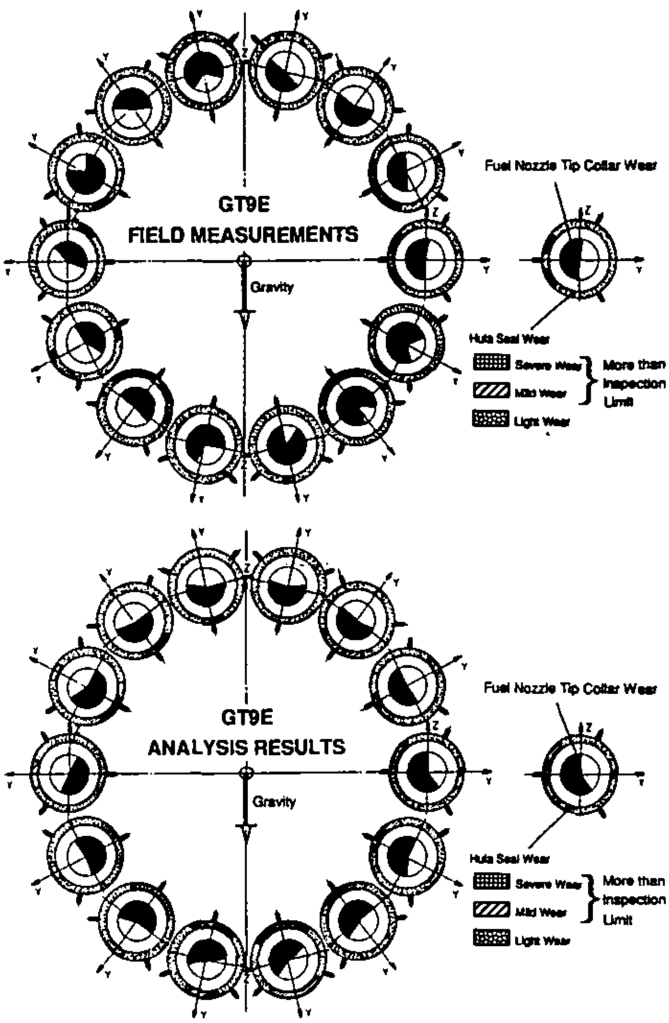


Figure 12. Field Measurements and Analytical Results for the MS9E

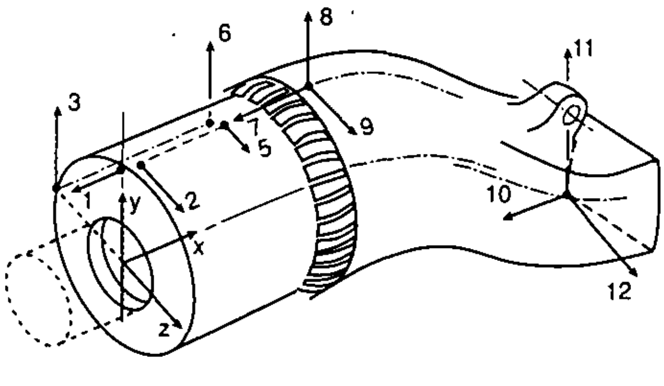


Figure 13. Accelerometer Locations on the MS9E-U Test Combustor

(as before with the MS9E). In most cases the net wear was *less* than in the MS9E case. It should be

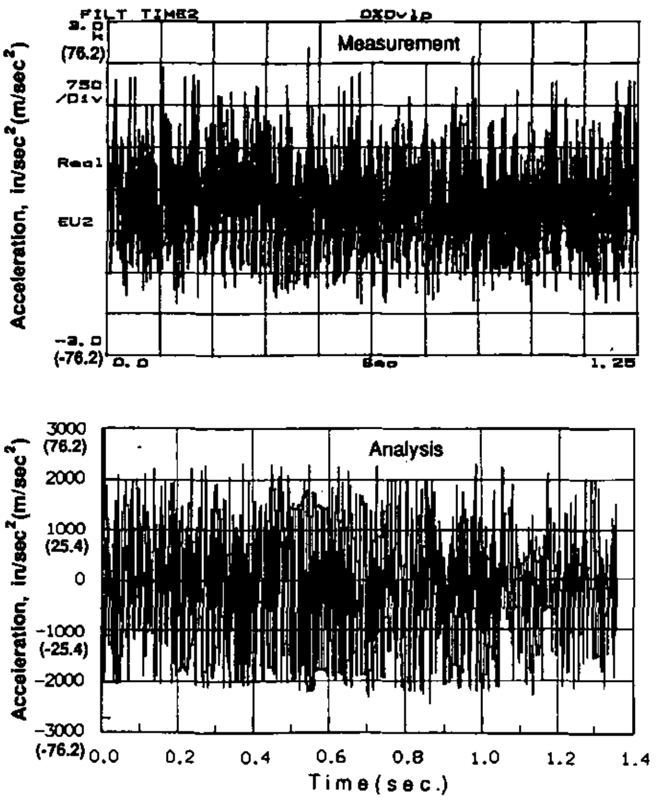


Figure 14. Test and Analysis Results at the Accelerometer Location #1 (on the Liner)

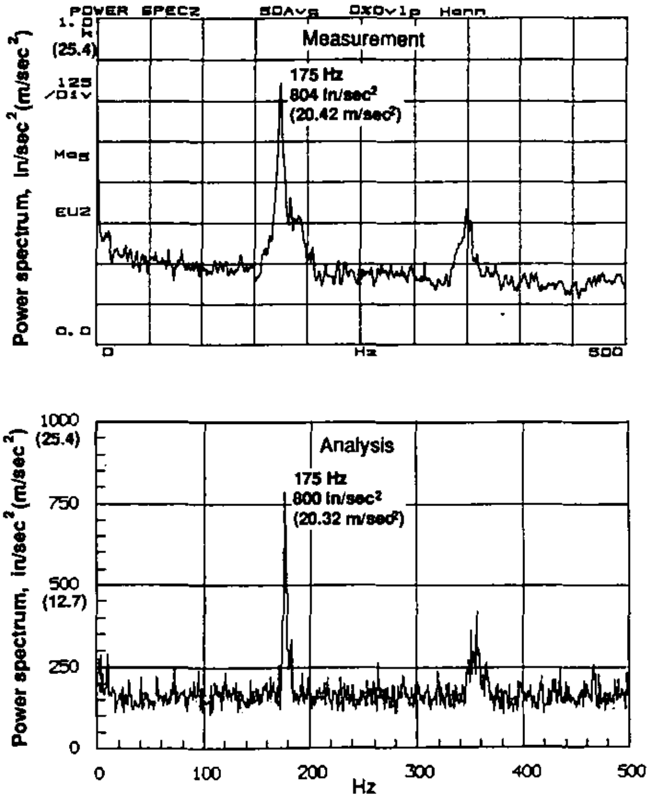


Figure 15. Test and Analysis Results at the Accelerometer Location #1 (on the Liner)

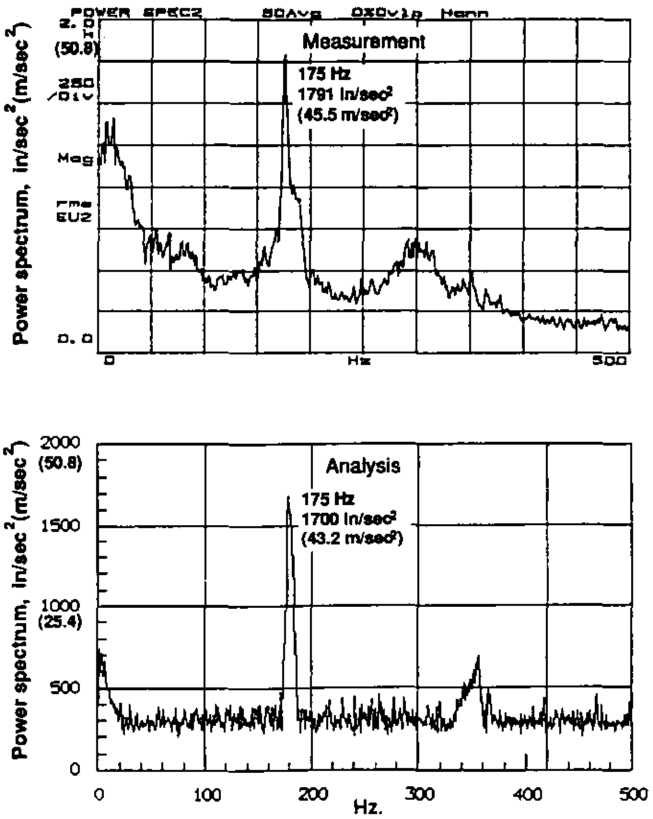


Figure 16. Test and Analysis Results at the Accelerometer Location #8 (on the Transition Piece)

noted that the loads were higher because of the heavier components, but relative rubbing velocities are smaller. The table in Figure 17 is a comparison of the dynamics and wear of the MS9E and MS9E-U designs. Clearly, the MS9E-U design allows for at least 30% longer inspection intervals (perhaps, over 11,000 hours compared to 8000 hours currently).

5. DISCUSSION AND CONCLUSION

The dynamic behavior of the combustion system has been analytically simulated with *MAP* which correlates favorably to laboratory acceleration measurements. Using these calculated loads and velocities along with measured material wear coefficients, component wear quantities were predicted with Archard's equation which are in good agreement with field data. A new combustion system design for the MS9E-U has been shown to have superior wear characteristics using *MAP* and Archard's equation. This methodology is now being employed to develop a reduced wear combustion system.

There are two different components of wear: Material based parameters such as hardness, wear coefficient, and system dynamic based parameters such as load and rubbing velocity. The wear can be reduced by changing these parameters. In the first group of

ELEMENT;			MS9E Power Loss (L,V)	MS9E Power Loss (L*V)	MS9E-Uprate Power Loss (L,V)	MS9E-Uprate Power Loss (L*V)	CHANGES	CONCLUSION	
TP;		Mass, lbs (kg)	82 (37)		117 (53)		+35 (16)	+43%	Heavier
Liner;		Mass, lbs (kg)	44 (20)		69 (31)		+23 (10.5)	+52%	TP.
Nozzle-Liner Contact		Load lbs (N) Velocity, in/s (cm/s)	36.5 (162.6) 0.091 (0.23)	3.3 (37.3)	42.5 (189.3) 0.068 (0.17)	2.87 (32.5)	+8 (26.7) -0.022 (0.06)	-14%	Less Wear
Hula-Sesi Contact;	Side;	Load, lbs (N) Velocity, in/s (cm/s)	30.3 (134.9) 0.096 (0.24)	2.9 (32.8)	30.5 (135.8) 0.067(0.17)	2.03 (23)	+0.2 (0.9) -0.029 (0.07)	-30%	Less Wear
	Top;	Load, lbs (N) Velocity, in/s (cm/s)	24.8 (110.4) 0.028 (0.07)	0.68 (7.7)	24.4 (108.7) 0.018 (0.04)	0.44 (5.0)	-0.4 (1.8) -0.010 (0.02)	-35%	Less Wear
	Bottom;	Load, lbs (N) Velocity, in/s (cm/s)	35.8 (159.4) 0.026 (0.07)	0.91 (10.3)	36.3 (161.7) 0.017 (0.04)	0.62 (7.0)	+0.5 (2.2) -0.009 (0.02)	-32%	Less Wear
Floating-Seal Contact;		Load, lbs (N) Velocity, in/s (cm/s)	17.0 (75.7) 0.073 (0.18)	1.23 (13.9)	14.5 (64.6) 0.058 (0.15)	0.84 (9.5)	-2.5 (11.1) -0.015 (0.04)	-32%	Less Wear

Figure 17. Comparision of the MS9001E and MS9001E-Uprate Combustors

parameters, new types of materials, heat treatment, coating could reduce the wear.

In the second group, one could work on the system dynamic to perform design sensitivity study and redesign the whole system for minimum wear and extended life if the wear is:

- **Load driven**, ie., where rubbing velocities are of small amplitude and low frequency, and contact loads are high, and where **design** changes such as location of liner stops would be appropriate, or,
- **Velocity driven**, ie., where rubbing velocities are harmonic, of high amplitude and frequency, and with moderate contact loads. Here, changes in **system** or **design** variables such as mass, inertia, CG location, spring-constant, etc., would yield substantial reduction in wear.

One could achieve long-life, low-wear designs via automated optimization procedures which are briefly introduced in Figure 4 as an evaluation methodology of wear and life.

Numerous design changes have been investigated to develop a reduced wear combustion system. For example, liner stop locations and clearances, liner and TP center of gravity locations, hula seal shape and stiffness, TP support stiffness, and floating seal configurations are being investigated to improve wear. Changes to the liner stops effect the entire system behavior; reducing the load times velocity $(L*V)$ quantity in Archard's equation by 15 to 20% at each contact area. The methodology has also allowed evaluation of unsuccessful redesigns and thereby eliminated costly and time consuming prototype testing. Combustion design engineers have gained insight into the dynamic behavior of the combustion system which will be invaluable in future developments.

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